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Bethesda, Md. 20084

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AN INVESTIGATION OF THE TOWING STABILITY OF THE HYTOW BODY

Shelton M. Gay, Jr.

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SHIP PERFORMANCE DEPARTMENT
DEPARTMENTAL REPORT

December 1979 DTNSRDC/SPD 0916-01

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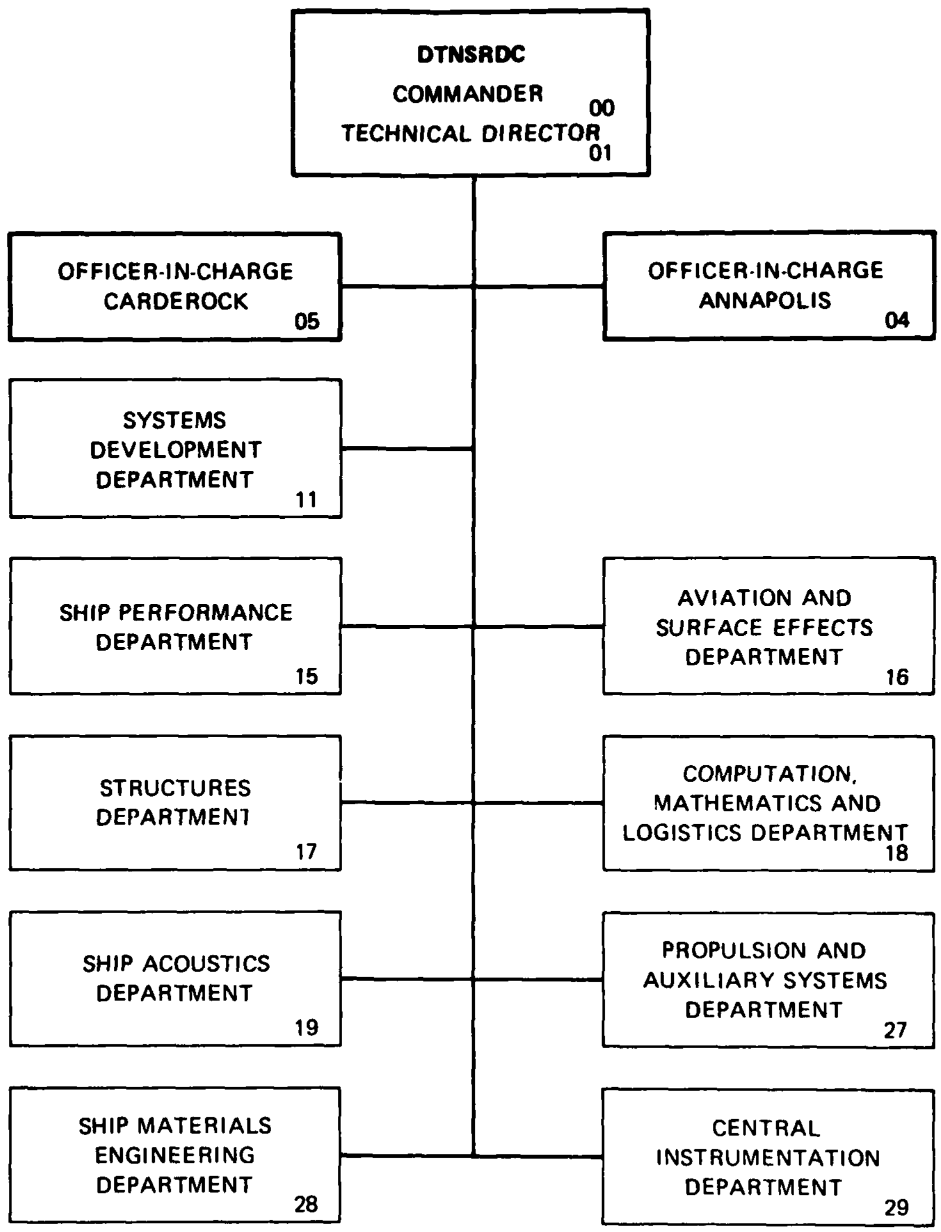

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A 9-foot long towed sonar housing, referred to as the "HYTOW", evidenced towing instability in two instances when towed at speeds near 30 knots from the hydrofoil patrol craft PCH-1. Towing experiments were conducted with a 1/5-scale model in the towing tanks at the David W. Taylor Naval Ship Research and Development Center to find a solution for the stability problem. An effective remedy consisted of flaring the body lines aft of the 70-percent length

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station into a truncated cylindrical shape that is terminated by a cylindrical cap having a diameter 50-percent of the maximum body diameter. It also was found that when so modified, the body towed more stably with the upper V-shaped horizontal stabilizer interchanged with the lower stabilizer and mounted inverted. With the recommended configuration changes the body towed stably (in model scale) to the full-scale equivalent speed of 45 knots.

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NOTATION

A	-	distance from towpoint to center of weight in water
a	-	the ratio A_2/A_1, also longitudinal location of W relative to the towpoint
B	-	buoyancy force
C_p	-	prismatic coefficient
C_x	-	coefficient with the type defined by subscript x
D	-	maximum diameter of body
F	-	force, also fineness ratio, L/D
f	-	force ratio
H	-	moment (torque)
I	-	moment of inertia
k	-	added mass factor
L	-	body length, also rolling moment
ℓ	-	ratio of body lengths
M	-	$\underline{M}/\rho V$
$\underline{M}$	-	total mass
M_A	-	added (hydrodynamic) mass
m	-	the ratio $\underline{M}_2/\underline{M}_1$
N	-	radius of gyration and yawing moment
N_0	-	yaw moment due to asymmetry
n	-	ratio of radii of gyration
q	-	dynamic head
R	-	fluid resistance or body radius
R_0	-	nose radius
R_1	-	tail radius
r_0	-	nondimensional nose radius
r_1	-	nondimensional tail radius
S	-	area
s	-	ratio of areas
T	-	time
t	-	ratio of times

U	-	towing speed
u	-	velocity ratio
$\mathbf{V}$	-	volume
$\mathbf{v}$	-	ratio of volumes
W	-	weight in water
w	-	the ratio W_2/W_1
x	-	generalized space coordinate
$\ddot{\overline{X}}$	-	$d^2\overline{X}/dT^2$, acceleration of point whose position is $\overline{X}$
Y	-	side force
Y_0	-	side force due to asymmetry
Z	-	angle of line tangent to body velocity vector in space coordinate frame
z	-	ratio of two values of Z
α	-	angle of attack
β	-	ratio of space coordinates
Δ	-	mean (average) apparent specific gravity of a body immersed in water
δ	-	the ratio Δ_2/Δ_1
Θ	-	angle of a reference line in the body relative to the frame of space coordinates
θ	-	ratio of two values for Θ
λ	-	scaling factor for linear dimensions
ρ	-	density of water
σ	-	angle defined by direction line from body towpoint to center of weight in water and a reference line in the body
ϕ	-	roll angle
ψ	-	yaw angle

ABSTRACT

A 9-foot long towed sonar housing, referred to as the "HYTOW", evidenced towing instability in two instances when towed at speeds near 30 knots from the hydrofoil patrol craft PCH-1. Towing experiments were conducted with a 1/5-scale model in the towing tanks at the David W. Taylor Naval Ship Research and Development Center to find a solution for the stability problem. An effective remedy consisted of flaring the body lines aft of the 70-percent length station into a truncated cylindrical shape that is terminated by a cylindrical cap having a diameter 50-percent of the maximum body diameter. It also was found that when so modified, the body towed more stably with the upper V-shaped horizontal stabilizer interchanged with the lower stabilizer and mounted inverted. With the recommended configuration changes the body towed stably (in model scale) to the full-scale equivalent speed of 45 knots.

ADMINISTRATIVE INFORMATION

This work was sponsored by NAVSEA 0322 under Program Element 63567N, Task 01700, David W. Taylor Naval Ship Research and Development Center Work Unit 1150-901-42.

INTRODUCTION

Two successive losses of a sonar body towed from the hydrofoil U.S.S. HIGH POINT (PCH-1) occurred in Puget Sound, Washington, in March and August 1978. The bodies were being tested for suitability for high speed towing as part of a program sponsored jointly by the United States and Canada.

The first body was lost as a result of body forces exceeding the strength of a weak link at the body towpoint. Damaged faired towline in the vicinity of the body suggested that it had struck a submerged object. The second loss resulted from broaching of the body while towed at a speed just slightly greater than 30 knots. The broached body presented a high resistance and the weak link failed as designed. Roll and pitch data indicate that both bodies experienced significant unsteadiness in roll and pitch when near the 30-knot towing speed. As a result of the losses, the United States, with the cooperation of the Canadians, undertook investigations to determine the causes and develop a solution. A 1/5-scale model of the body provided by the Canadians was evaluated in the deep-water basin at the David W. Taylor Naval Ship Research and Development Center (DTNSRDC) to determine the likely cause of the broaching and to develop acceptable remedies.

This report contains a summary of the evaluations conducted, the remedial actions taken, and recommendations for modification of the prototype body.

ANALYSIS OF LOSSES

Recordings of roll, pitch and cable tension taken just prior to the losses of the prototype bodies in both cases indicated a high degree of unsteadiness in roll and a smaller degree of unsteadiness in pitch. The latter also indicated its presence by fluctuations in the towline tension. Since roll and sway are coupled by the cable constraint, the roll was accompanied by a varying tow-off or kite angle.

The roll, in particular, was characterized by a quasi-periodic variation in which a random low frequency component could be distinguished with a much higher frequency random component superimposed on the lower frequency trace. The frequency separation of the two components was on the order of two decades. The towing characteristics of

the body also were characterized by relatively large shifts in the mean value of the roll angle at different speeds.

The towing characteristics described above are symptomatic of a body subject to turbulent flow separation. The frequency spectra of the forces produced by such flows are typically broadband, with significant energy in the low frequency end of the spectrum. Moreover, the frequencies of the low frequency spectral energy peaks tend to shift with the Reynolds number of the flow.[1] The HYTOW body has a fineness ratio of three, hence flow separation is to be expected.

On the other hand, the instrumentation records and eyewitness accounts of events preceding the loss of the second body gave strong indication of the on-set of a classic example of a lateral instability. An isometric view of the track of the second towed body during the period immediately preceding its broaching and subsequent loss is shown in Figure 1. The extreme maneuver could be caused by either a large asymmetrical shift of the boundary of the region of separated flow or the onset of true lateral instability.

Three bodies geometrically similar to the HYTOW were towed by the National Research Council[2] (NRC) of Canada to determine their stability characteristics. The HYTOW body constitutes a fourth in the series. The results of these towing trials were examined to determine if inferences regarding the stability of the HYTOW body could be gleaned from the earlier results.

For this purpose the requirements for dynamic similarity of towed systems were analyzed in detail and are reported in Appendix A. Briefly, the results indicate that for submerged bodies complete dynamic similarity will exist only if the ratio of their weights in water is equal to the cube of the ratios of corresponding lengths, i.e.,

$$W_2/W_1 = (L_2/L_1)^3 = \lambda^3 \qquad [1]$$

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[1]A complete list of references is given on page 44.

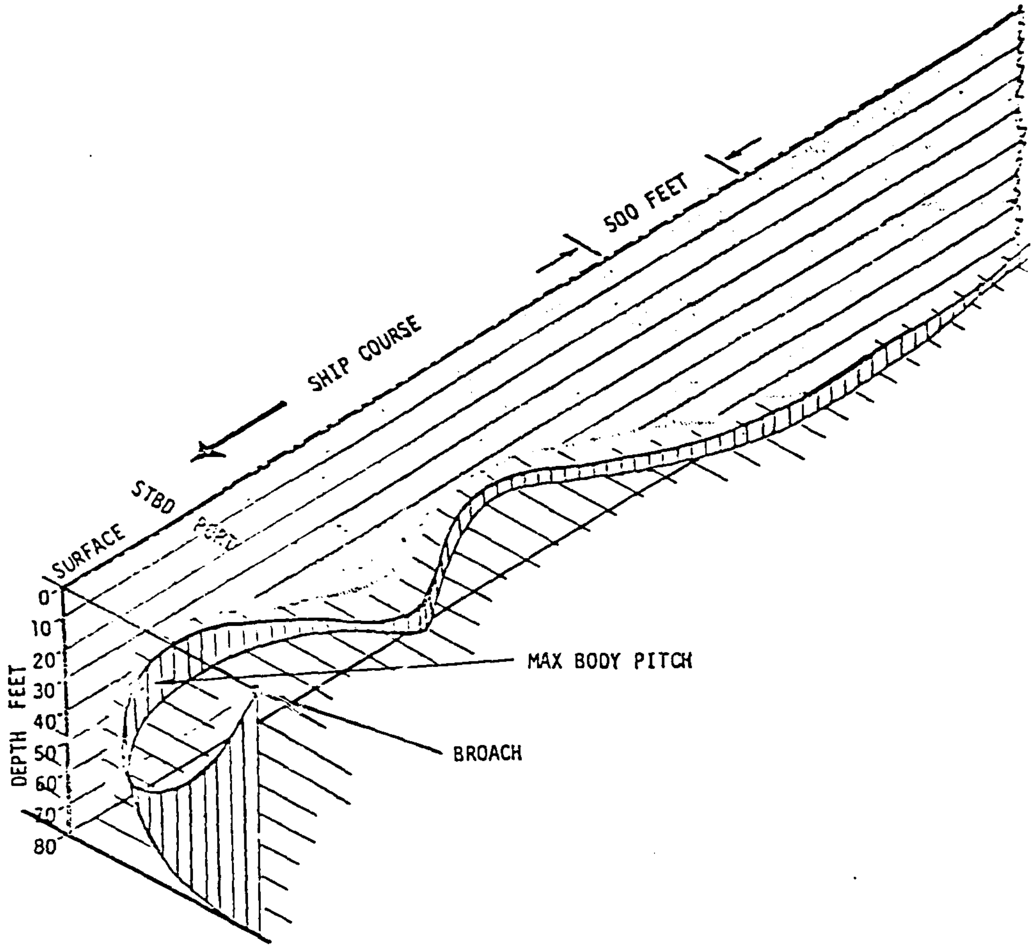

Figure 1 - Isometric Representation of HYTOW Body Track Prior to Loss

(from Boeing Co. Ltr H-7308-1000-2105 to C. Ray, dtd 8 Aug 78)

where W is the weight in water and L is a linear dimension. Unfortunately, the weight-in-water of the bodies tested by NRC did not meet the above criteria.

In Appendix B, an equation is derived for the steady-state roll angle resulting from an asymmetrical force and moment acting on a towed body. The effect of asymmetrical flow separation, for example, is equivalent to a built-in asymmetry. If the flow patterns are similar the towing speed at which the static roll angle increases to the value required for broaching should be predictable from model test results. Both prototype and model Reynolds number are sufficiently large to ensure the existence of turbulent flow.

The results for dynamic similarity and equal static roll angles are the same (equations A-11 and B-8), namely

$$u = U_2/U_1 = \sqrt{\frac{W_2/W_1}{\lambda^2}} \qquad [2]$$

The limiting speed for stable tow of the various models was computed using the body first listed in Table 1 as the model and each succeeding body in the list as the prototype. The pertinent information and results are listed in Table 1.

We note good agreement for the 1.8-ft (0.55 m) and 9.0-ft (2.7 m) bodies but not the 4.5-ft (1.4 m) long body. It was noted, however, in the NRC report[2] that the 4.5-ft (1.4 m) body was not tested to its limiting speed as it was "very unstable" at 26 knots.

Unfortunately the results of the analyses fail to provide a basis for determining whether the undesirable towing characteristics are due to a flow separation or a classic lateral instability.

TABLE 1 - SUMMARY OF MODEL TESTS

Model Length	Weight in Air	Weight in Water	Cable Scope	Max. Stable Speed (knots)	
ft(m)	lb(N)	lb(N)	ft(m)	Observed	Predicted
1.8(0.55)	98(435)	85(378)	56(17)	36	--
1.8(0.55)	68(302)	55(245)	56(17)	26	29
4.5(1.4)	880(3914)	760(3380)	86(26)	25½	43
9.0(2.7)	2369(10537)	1540(6850)	100(30)	31-32	31

DESCRIPTION OF THE PROTOTYPE TOWED BODY AND MODEL

The towed sonar housing is a fin-stabilized body of revolution. It is towed from a point on its upper surface located 40% of body length L aft of the nose, a position that coincides with the position of the maximum diameter D. The basic body shape is derived from the David W. Taylor Model Basin* (DTMB) Series 58[3] form. In the notation descriptive of the Series 58 forms it would be designated:

58 --	40 --	50 --	10 --	65 --	03
Series	Position of max. thickness in % of length, m	Non-dimensional nose radius in %, r_o	Non-dimensional tail radius in %, r_1	Prismatic Coefficient in %, C_p	Fineness Ratio, λ

These quantities are defined in Table 2.

The overall configuration of the prototype body and dimensions of the stabilizers are shown in Figure 2. The upper horizontal fin had a dihedral angle of 26 degrees. The stabilizers are attached to four longitudinal skegs** also illustrated in Figure 2. A model, geometrically similar to the prototype body, was supplied by the Defense Research Establishment-Atlantic (DREA) of Canada. The model was 1.8-ft (0.55 m) long and constructed of fiberglass. The scale ratio of prototype to model was five. The physical characteristics of the prototype body and the model are presented in Table 2.

The full-scale towcable is 0.865 in. (2.2 cm) in diameter and is double-armored. It is faired with a Fathom Oceanology Ltd. Flexnose fairing with a chord length of approximately 4 in. (10 cm) and a thickness of 1.05 in. (2.7 cm). The attachment of the towcable to the body

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*Now the David W. Taylor Naval Ship Research and Development Center.

**Used here to mean a very low aspect ratio stabilizing fin due to the similarity with the skeg of a surface ship.

TABLE 2 - PHYSICAL CHARACTERISTICS OF PROTOTYPE BODY AND MODEL

	Prototype	Model
Total Length (L)	9 ft (2.7 m)	1.8 ft (1.55 m)
Maximum Diameter (D)	3 ft (0.9 m)	0.6 ft (0.2 m)
Nose Radius (R_o)	6 in. (15 cm)	1.2 in. (3 cm)
Tail Radius (R_1)	1.2 in. (3.1 cm)	0.2 in. (0.6 cm)
Prismatic Coefficient ($C_p = \forall/\pi D^2 L/4$)	0.65	0.65
Volume ($\forall$)	41.4 ft^3 (1.2 m^3)	0.33 ft^3 (.0096 m^3)
Weight in Air (W_a)	2369 lb (10538 N)	19 lb (84 N)
Weight in Water (W_w)	1540 lb (6850 N)	12.3 lb (54.8 N)
Static Trim in Pitch and Roll (deg)	0	0
Moments of Inertia	Unknown	Unknown
Nondimensional Nose Radius ($r_o = R_o L/D^2$) x 100	50	50
Nondimensional Tail Radius ($r_1 = R_1 L/D^2$) x 100	10	10

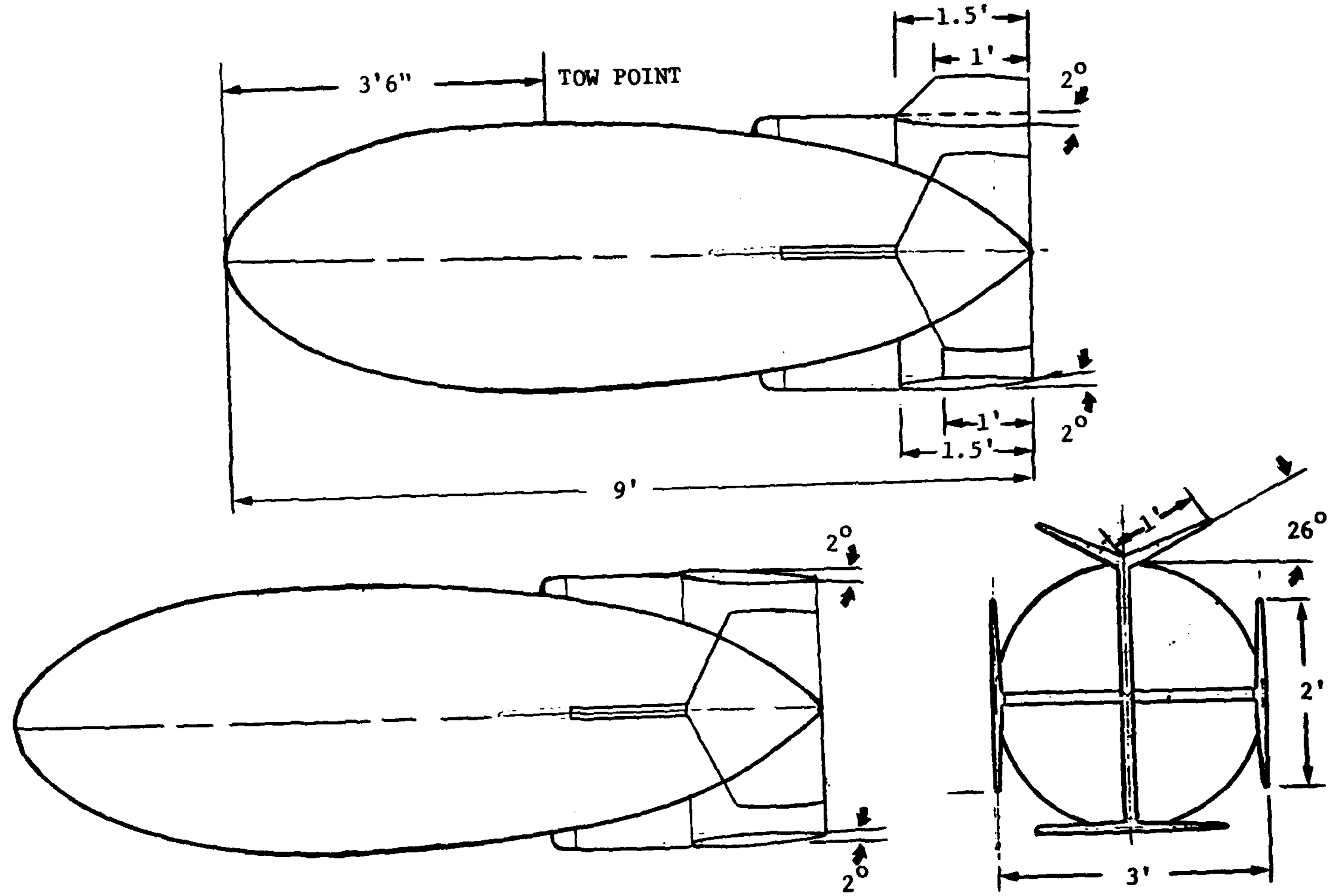


Figure 2 - The Full-Scale HYTOW Body

is made by means of a ball socket intended to decouple the body and towcable in yaw, roll, and pitch. During the trials the body was towed by a 100-ft (30 m) length of towline.

To simulate prototype motions, the model (on the basis of Froude scaling) should have the physical characteristics listed in Table 2. The towline should ideally be scaled as shown below:

Thickness of model fairing	=	0.21 in. (0.53 cm)
Chord of model fairing	=	0.8 in. (2 cm)
Scope of towline	=	20 ft (6.1 m)

The model towline was a 3/16 in. (0.48 cm), three conductor, armored cable faired with a rubber trailing fairing 0.22 in. (0.56 cm) thick and 0.9 in. (2.3 cm) in chord length. The fairing was attached by clips to the towcable on 2 in. (5 cm) centers. The total length of cable was 20.75 ft (6.3 m) of which the 15 ft (4.6 m) section nearest the body was faired. The upper end of the cable was attached to the carriage 3.75 ft (1.1 m) above the water surface so that a cable length of 17 ft (5.2 m) was submerged. It was assumed that the moments of inertia of the model would be reasonably close to values scaled from the prototype if the body were ballasted to achieve the scaled weight in water and the proper trim.

EXPERIMENTAL APPROACH

Two approaches were available for investigation of the flow separation: use of a constrained model to obtain measurements for development of force and moment spectra; or use of a towed model with measurements of the motions to indicate the effectiveness of flow-control devices. The latter approach was adopted as being most direct and economical. Determination of the spectra of the motions was not necessary as increased steadiness of the towed body indicated the effectiveness of the flow control devices.

The lack of data regarding the effectiveness of flow control devices on bodies as bluff as the HYTOW mandated a heurestic approach. A variety of flow-control devices developed to alleviate flow separation problems in the aeronautical industry was investigated. Since a successful device must be suitable for practical implementation, it was necessary that it not be susceptible to handling damage, not generate vortices with a shedding frequency near the sonar operating frequency and not require powered motors as would be needed for active boundary layer control by blowing through slots or by suction.

A very simple experimental approach therefore was indicated: design and evaluate a series of flow-control devices until a suitable device was found.

REYNOLDS NUMBER EFFECTS

The Reynolds number based on body length for the prototype and model bodies at equivalent Froude-scaled speeds are compared below:

	Towing Speed (knots)	R_n
Prototype	30	3.9×10^7
Model	13.4	3.7×10^6

It is evident that in the range of speeds of interest the flow over the model will be turbulent. However, in view of the highly favorable pressure gradient over the forward 40% of the model length, it is likely that the flow there will remain laminar even in the presence of roughness. Therefore, trip rings were designed for installation at the 30% length station on the model to increase the longitudinal extent of turbulence and to thicken the turbulent boundary layer to more closely simulate the flow over the prototype. Two rings were designed; one 3/8 in. (0.9 cm) diameter and the other 1/8 in. (0.32 cm) diameter.

APPENDAGES

Several appendages were designed and evaluated for effectiveness on the 1/5-scale model. These consisted of flow control/modification devices and motion damping and stabilizing appendages.

FLOW CONTROL DEVICES

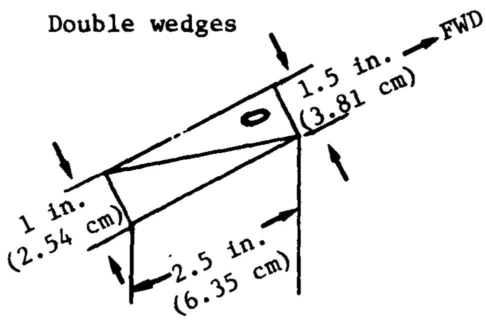
The flow control devices consisted basically of types that energize the boundary layer by mixing high velocity water with the lower-velocity boundary fluid and types that stabilize the flow separation boundary. The first we shall call "flow mixers" and the latter "boundary fixers". The flow mixing devices are listed in Table 3.

The boundary fixing devices consisted of segments of truncated cylinders mounted between the skegs that served to flare the afterbody lines such that the body is terminated with a circular flat surface normal to the axis of symmetry. The devices have been designated as "sleeves" because they were designed to slip over the after part of the body. The first sleeve was a crude, untapered, wooden affair designed to evaluate the concept. Two other sets were subsequently evaluated. The dimensions of the sleeves and attachment locations on the body are shown in Figure 3. For convenience of installation, the sleeves were split horizontally so that either the top or bottom half could be installed independently of the other.

DAMPING AND STABILIZING APPENDAGES

The damping appendages consisted of two different sizes of bow-mounted fins, as illustrated in Figure 4. The fins were mounted in a bracket curved to fit the body contour so that the gap between the body and the after part of the fin was closed by the bracket. The position of the fins mounted on the body is shown in Figure 5. For brevity we shall refer to these fins as the "small fin" and "large fin".

TABLE 3 - FLOW MIXING DEVICES

Item	Description	Location of Model
Ramps	Double wedges 	4 ramps spaced equally circumferentially with leading edge at 75% L (see Figure 5)
Vortex Generators	Cylinders 0.375 in. (0.95 cm) diameter x 0.375 in. (0.95 cm) high mounted with axis of symmetry normal to towed body surface	36 total (6 per quadrant) spaced approximately on 0.75 in. (1.9 cm) centers between the skegs at 75% L (see Figure 5)
Ring Foil	1.5 in. (3.8 cm) chord flat plate encircling the body at a height of 1 in. (2.5 cm) and an angle of inclination of 5^{o} leading edge up relative to local slope of the body	Mounted with 1/4-chord at 70% L (see Figure 5)

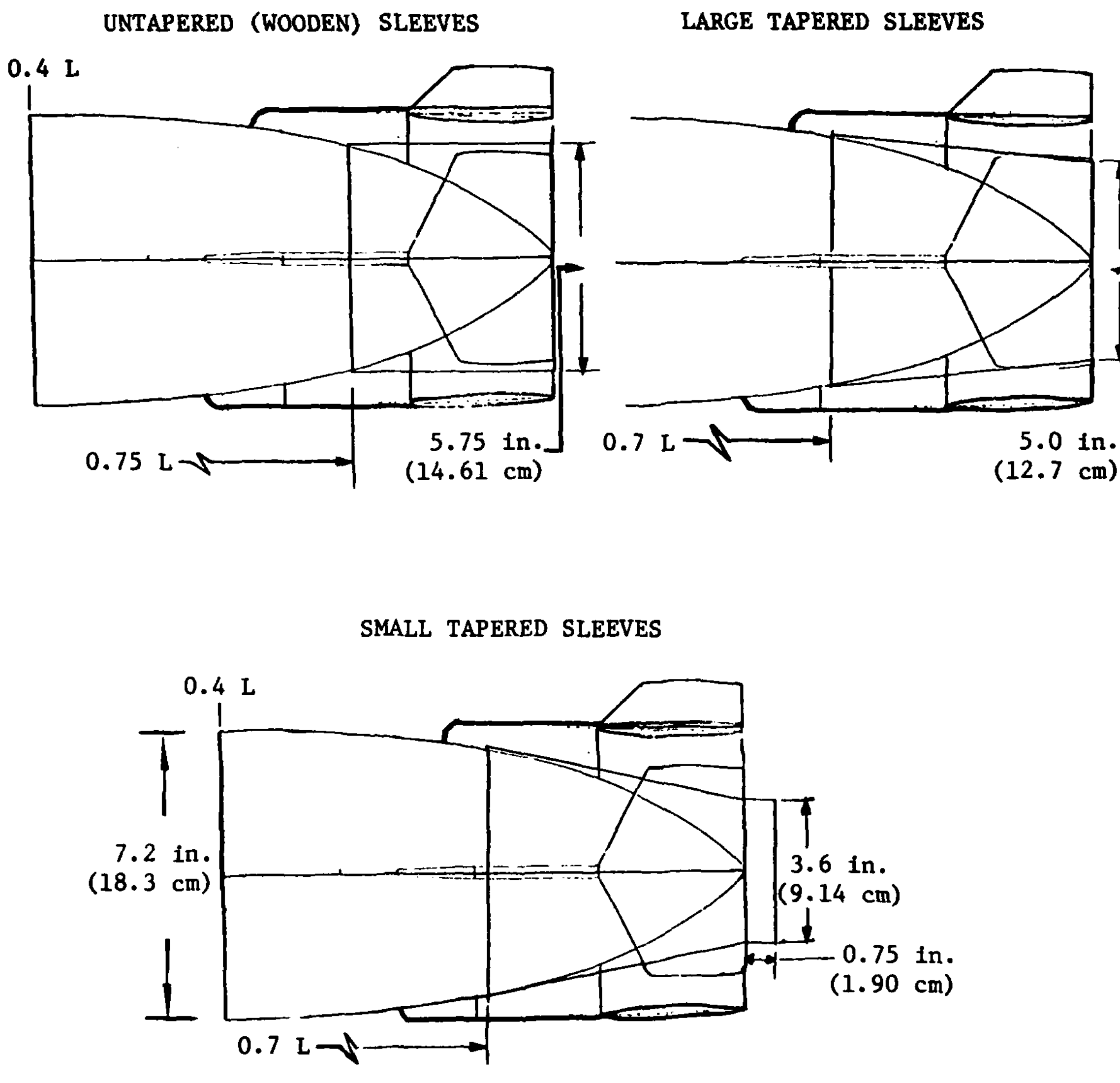
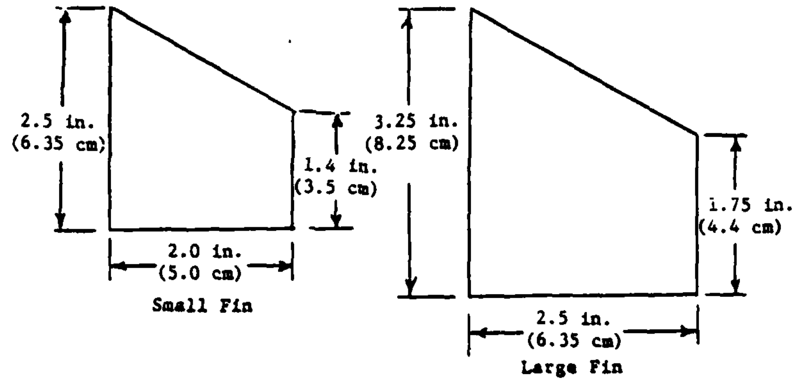
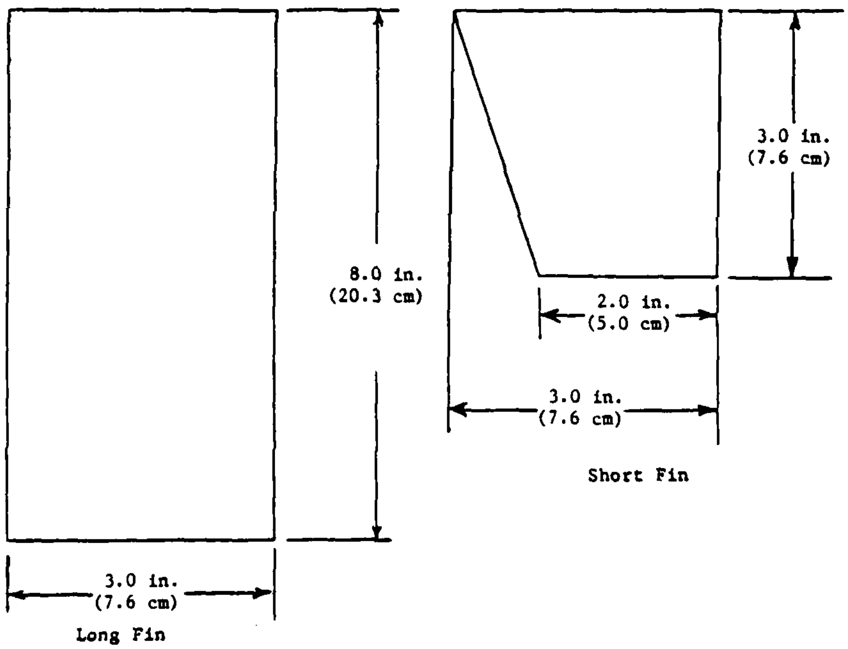


Figure 3 - Flow-Separation-Boundary Fixing Devices



DAMPING (BOW) FINS



STABILIZING FINS

Figure 4 - Damping and Stabilizing Appendages

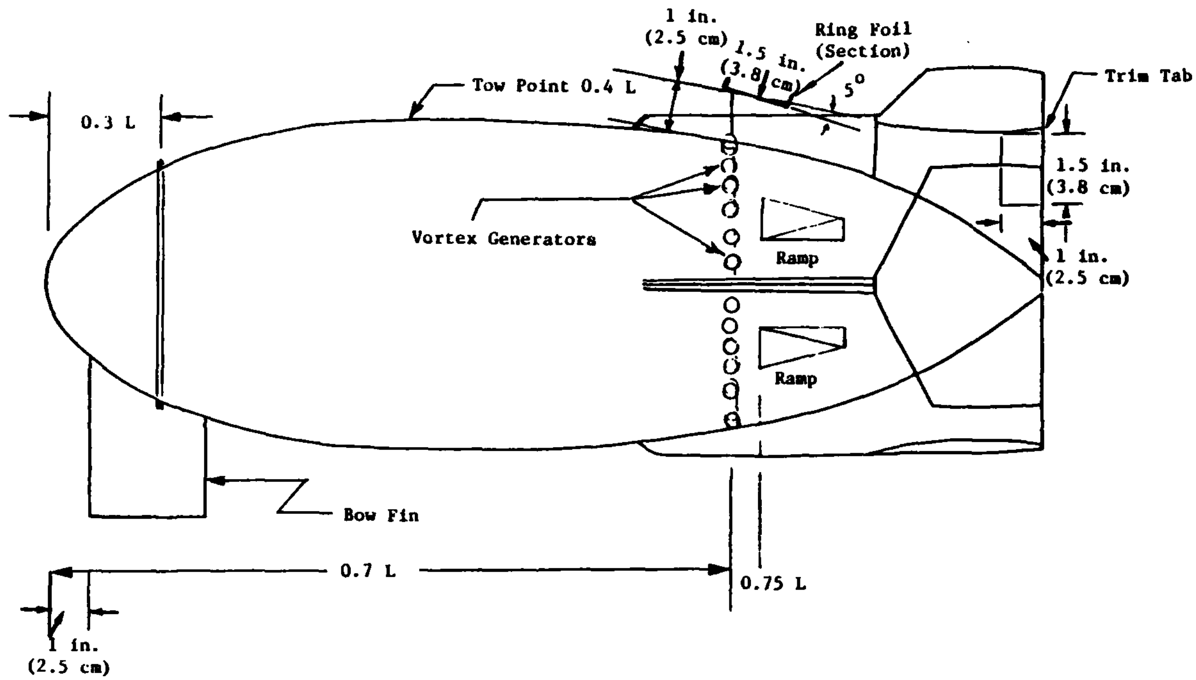


Figure 5 - Location of Attachments used for Instrumented Towing Trials

The stabilizing fins, also shown in Figure 4, were designed to mount on a bracket attached to the lower vertical skeg at the rear of the body. The two horizontal stabilizers were modified to mount on either the upper or lower skeg or to the lower end of the stabilizing appendage. Both the damping and stabilizing devices were placed on the underside of the body to provide a favorable coupling between side-slip and roll. Also, in that position, their presence would not interfere with a saddle which contacts the upper side of the body during launching and retrieval.

TRIM TABS

A trim tab was installed on the upper vertical skeg, just below the upper horizontal stabilizer, as illustrated in Figure 5. This position was selected because it produces the smallest adverse rolling moment per unit of yawing moment.

Horizontal trim tabs were installed in the form of two 30% mean-chord, half-span flaps. The use of half-span flaps permitted their installation on either horizontal stabilizer.

OTHER STABILIZING TECHNIQUES

In addition to the appendages, the model was ballasted in water to trim down by the nose between 5 and 8 degrees. Roll and sway are coupled parameters in a towed housing. Roll results primarily from a side force on the body that produces sway, which in turn, due to the towcable, induces a rolling couple. A body trimmed down by the nose thus develops a yawing moment that tends to turn the nose back toward the zero sway (and zero roll) position. (See Appendix B.)

INSTRUMENTATION

Body roll, pitch, and yaw rates were measured at the body. Tension was measured at the upper end of the towline. Not all measurements were taken for all conditions. Roll, however, was recorded during every run.

The roll and pitch measurements were obtained with Humphrey, Inc. Model CP-15 pendulum driven potentiometers. The yaw-rate transducer was a Humphrey, Inc. Model RT-01. A 100-lb, strain-gage load cell built by DTNSRDC was used to measure towline tension. All data were recorded on a Sanborn, six-channel, stripchart recorder.

EVALUATION PROCEDURES

The towing experiments were conducted from Carriage 2 in the deep-water towing basin at DTNSRDC. This basin is 22-ft deep, 55-ft wide, and provides a length-of-run of about 1800 ft. The carriage speed is controllable to within 0.1 knot and the top speed is 20 knots.

The final configurations evaluated are listed in Table 4. The first three configurations listed were for exploratory purposes, and visual observations only were made. The model was ballasted nose-down about 10° for these tows.

The model was towed at the same cable scope in all runs. Carriage acceleration was held nearly constant until the desired speed was reached. Measurements of the parameters were recorded throughout the run from initial acceleration to completion. Visual observation of the body was maintained during every run. This provided a basis for evaluating the records, as the visual observation was sufficient to determine the relative effectiveness of the configuration under test.

RESULTS

The body was towed initially without instrumentation (configurations 1 through 3). The observed motions were similar to those recorded for the prototype body. These initial tows also showed that the yaw-trimming tab had sufficient power to bias the towing position to either port or starboard.

TABLE 4 - HYTOW MODEL CONFIGURATIONS EVALUATED

Conf.	Trip Ring	Tapered Sleeves[a]		Ring Foil	Dihedral Fin Location		Pitch Trim Tabs	Stab. Fin[b]	Bow Fin[c]		Moveable Lower Horiz. Fin	Vert. Side Fins	Static Pitch (deg Nose Down)	Weight in Water (lb/N)
		Top	Bot.		Top (V)	Bot. (Λ)			Fin & Mount	Mount Only			(in water - Zero Speed)	
1[d]	L				x							on	nose down*	12½/56
2	L				x							on	nose down*	12½/56
3	L				x				(s)			on	nose down*	12½/56
4	L			x	x			(s)		x		on	7½	12-13/54-58
5[e]	L			x	x			(s)	(s)			on	7½	12-13/54-58
6	L			x	x			(s)	(s)			on	8½	12-13/54-58
7	L			x	x			(s)	(l)			on	8½	12-13/54-58
8	L			x	x			(s)	(l)			off	8½-9	12-13/54-58
9	L				x			(s)		x		on	10	12-13/54-58
10	L		L, wood		x			(s)		x		on	10	12-13/54-58
11	L		L, wood		x			(s)		x	x	on	10	12-13/54-58
12	L		L, wood		x			(l)		x	x	on	4	12-13/54-58
13	L	L, wood	L, wood		x			(s)		x	x	on	---	12-13/54-58
14	L	L, wood	L, wood		x		x	(s)		x	x	on	---	12-13/54-58
15	L		L, wood		x		x	(s)		x	x	on	6	12-13/54-58
16	L		L, wood		x		x	(s)			x	on	6	12-13/54-58
17	L		L, wood		x		x					on	7	12-13/54-58
18	L				x							on	---	12-13/54-58
19	L				x							on	0	13½/60
20					x							on	0	13½/60
21		S	S			x						on	7-8	12½/56
22	S	S	S			x						on	7-8	12½/56
23	S		S			x						on	7-8	12½/56
24			S			x						on	7-8	12½/56
25			L			x						on	6	12½/56
26	S	S	S			x						on	8	12½/56
27	S	S	S		x							on	7½	12½/56

a. L = large; S = small.
b. s = short fin; l = long fin.
c. s = small fin; l = large fin.
d. Used a 3/8-in. (0.95 cm) diameter vortex gen.
e. Used wooden wedges.
*Visual observations only were made.

The body was instrumented for the rest of the evaluations. The roll and yaw-rate outputs, supported by visual observations, provided the data necessary to judge the steadying effect of the various modifications. The pitch output provided zero-speed pitch trim information and, with the tension data, an indication of the relative effect on body drag of the various configurations evaluated. This inference was possible because the hydrodynamic stiffness in pitch was not changed during the evaluations and, since the towpoint was located on the upper surface of the body, an increase in drag at a given towing speed results in an increase in pitch. The towing tension likewise increases primarily from the higher hydrodynamic lift attending the larger angle of attack. The results of the instrumented tows are presented in the following paragraphs.

The unmodified body performance is shown in Figure 6. This corresponds to configuration 20 of Table 4. The body was ballasted to weigh 12-1/2 lb (56 N) in water (equivalent to 1540 lb (6850 N) full-scale) and with a static trim of zero degrees. The intensity of the unsteadiness increased with speed and is evident in the yaw rate, roll and tension data traces. The body towed with a nose down pitch between 5° and 6°. Only a very slight unsteadiness appeared in the pitch trace. The use of a boundary layer trip alone (located at 30% of the body length) had no perceptible effect on roll and sway motions. Under these conditions it was impossible to control the lateral motions (roll and sway) by means of the yaw trimming tab.

The only flow mixing device that gave improved towing characteristics was the ring foil when used with other stabilizing devices. The effect may be seen by comparing the towing characteristics of the model equipped with configurations 4 and 7. The appendages used with these configurations are listed below:

Figure 6 - Towline Tension, Roll, Pitch and Yaw Rate of the HYTOW Model in Prototype Configuration (Configuration 20)

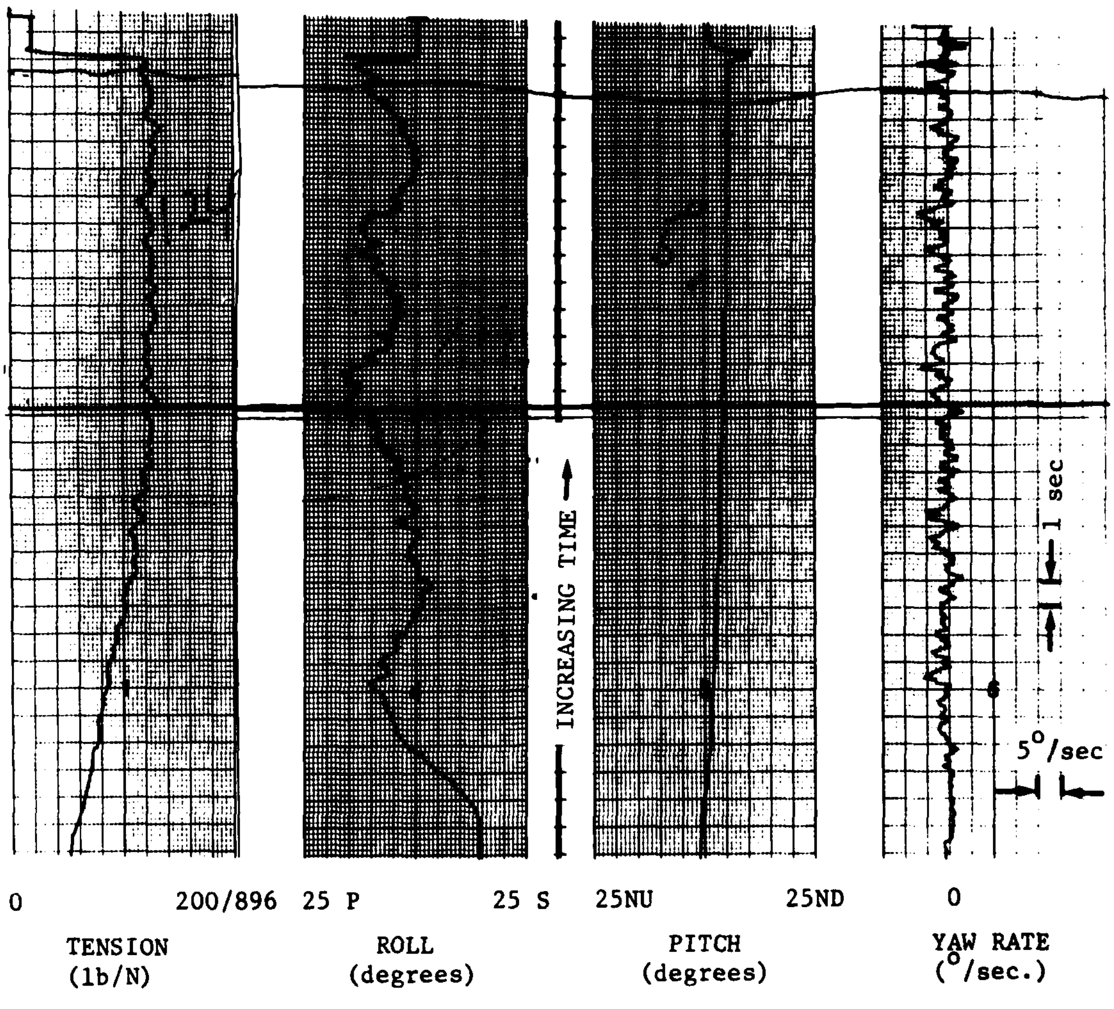


Figure 6a - At 20-Knot Towing Speed

Figure 6 (Continued)

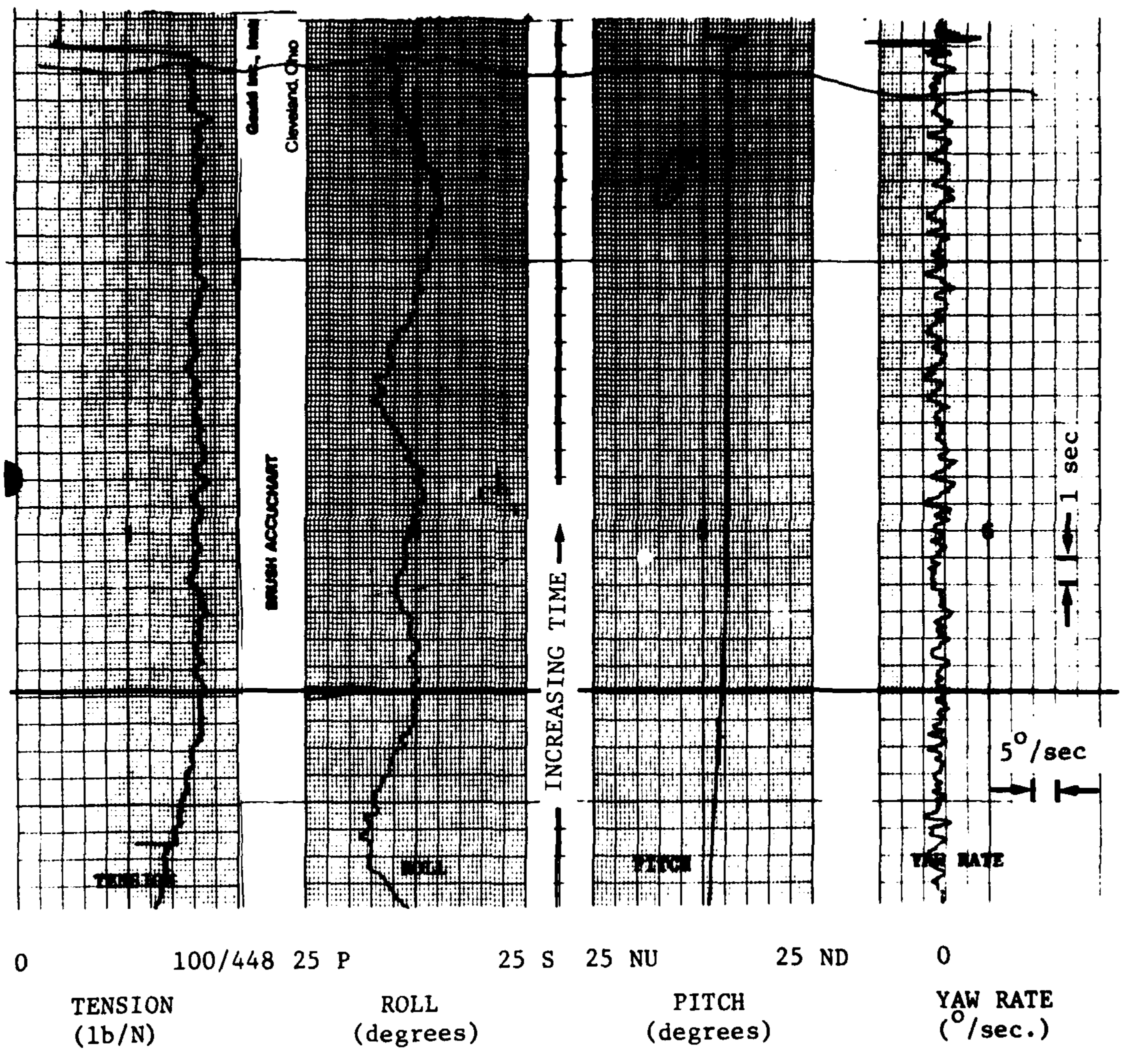


Figure 6b - At 15-Knot Towing Speed

Figure 6 (Continued)

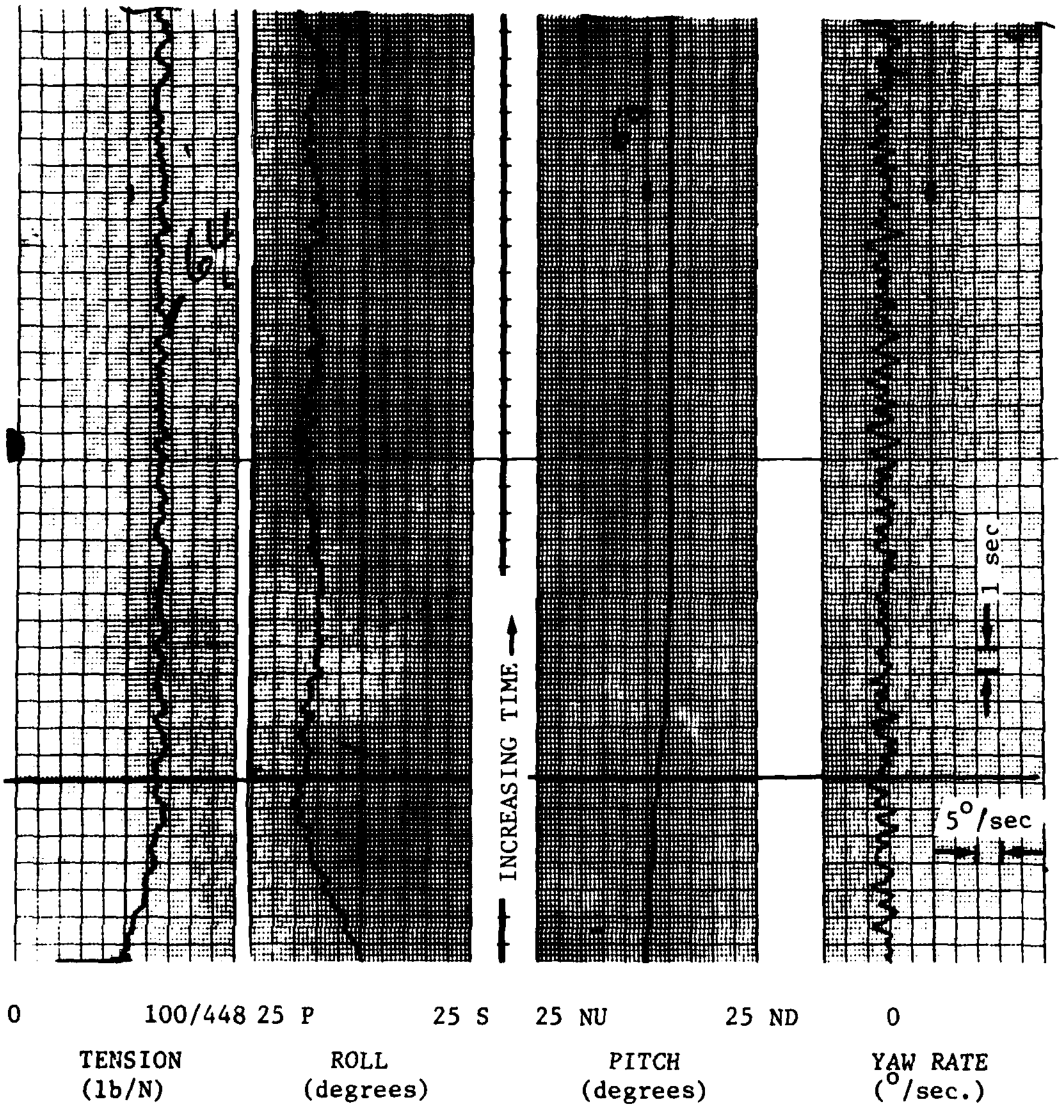


Figure 6c - At 13-Knot Towing Speed

Config.	Turb. Stim.	Ring Foil	Stabilizer Fin	Bow Fin	Static Pitch (deg)
4	x	x	Short	Mount	7-1/2
7	x	x	Short	Long	8-1/2

Towing results are compared at 18 knots in Figure 7. The recorder paper speed for configuration 4 was slower than for configuration 7 by a factor of 5. Even so, it is evident that configuration 4 is much less steady than configuration 7. The ring foil would not provide a satisfactory solution in any event, however, as it would be very vulnerable to damage during launch-retrieve evolutions.

The bow fin used alone (configuration 3) substantially reduced the unsteadiness in roll and yaw, but it was not possible to trim the body to tow dead-astern (zero roll and sway). Configuration 9, however, which consisted of the short vertical stabilizer fin extension (with the lower horizontal stabilizer mounted on the bottom of the extension) and boundary layer trip, also reduced the unsteadiness and proved possible to trim. The results at 13, 15, 18 and 20 knots are shown in Figure 8. The effect of flow separation is evident so this modification alone was not adequate. Nevertheless, it is evident from the results that the additional arrow stability and damping provided by the lower vertical fin extension is substantially beneficial.

At this stage of the evaluation, it was surmised that separation was occurring probably ahead of the 70% length station and that it was unlikely that the region of separation could be reduced to an acceptable size by introducing (mixing) higher velocity fluid into the boundary layer. It was therefore decided to attempt to fix the boundary of the separation region by expanding the body lines aft of about 70% L and terminating the expanded profile in a bluff circular base. To test the concept a set of sleeves was constructed, described as the wooden sleeves in Figure 3.

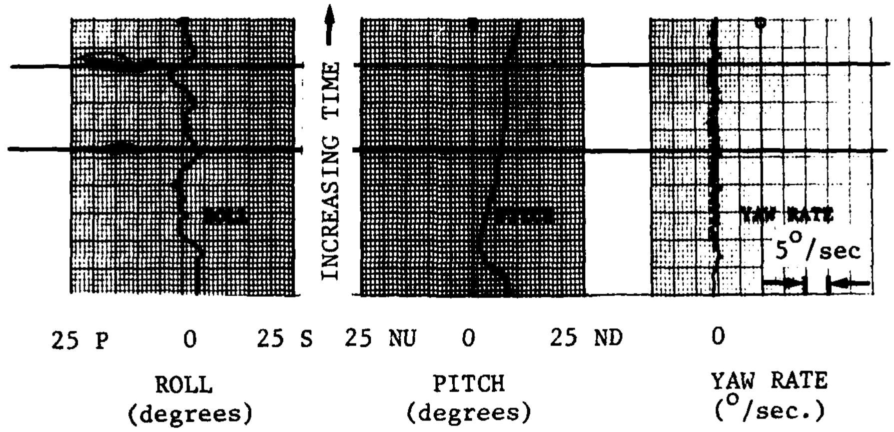


Figure 7a - Without Bow Fin (Configuration 4)

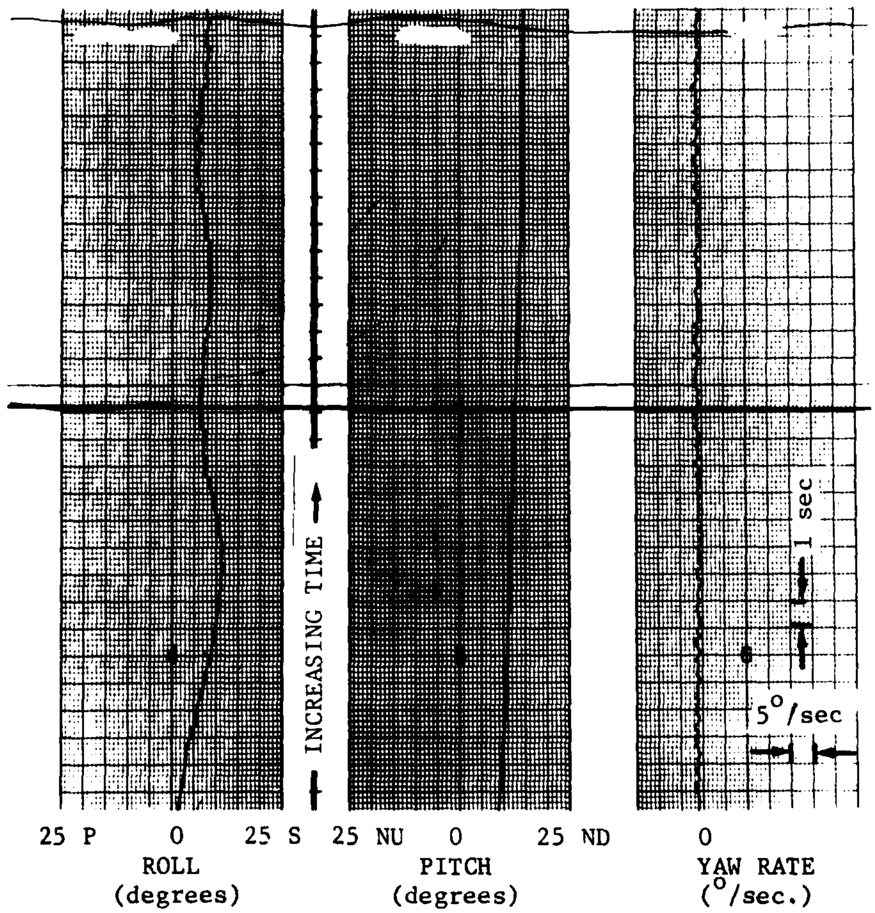


Figure 7b - With Short Bow Fin (Configuration 7)

Figure 7 - Comparison of HYTOW Towing Characteristics at 18 Knots with Ring Foil, Small Vertical Fin and Boundary Layer Trip

Figure 8 - HYTOW Towing Characteristics
with Short Vertical Fin and Boundary Layer Trip
(Configuration 9)

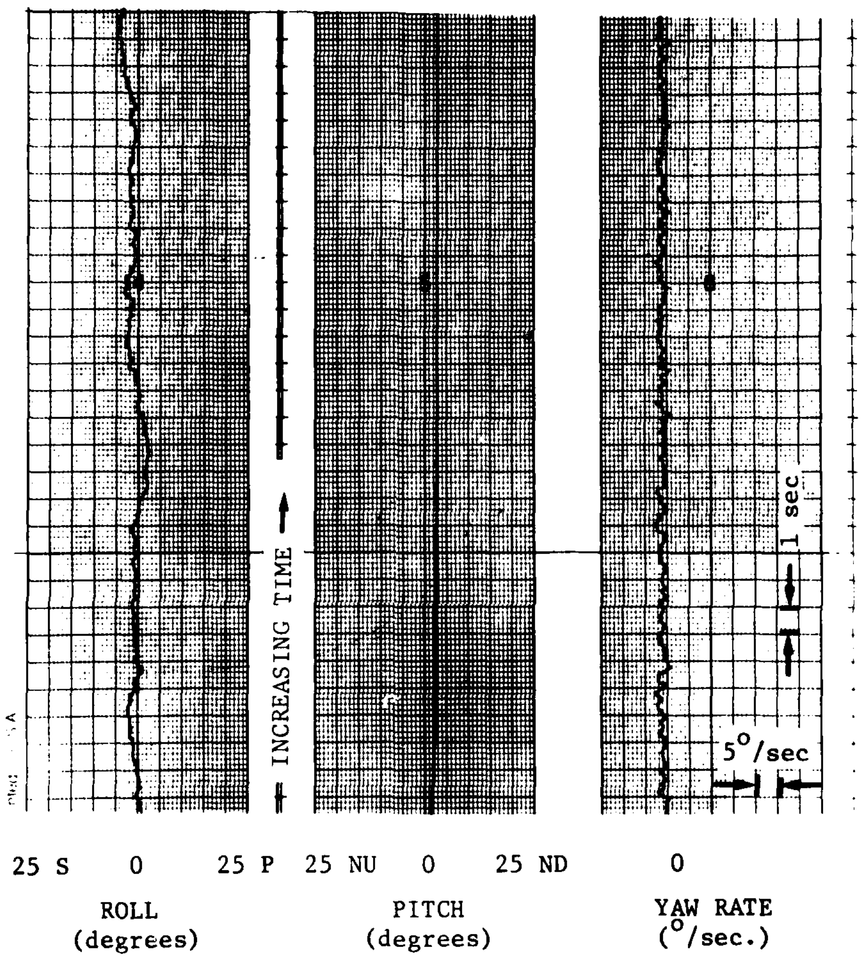


Figure 8a - At 13-Knot Towing Speed

Figure 8 (Continued)

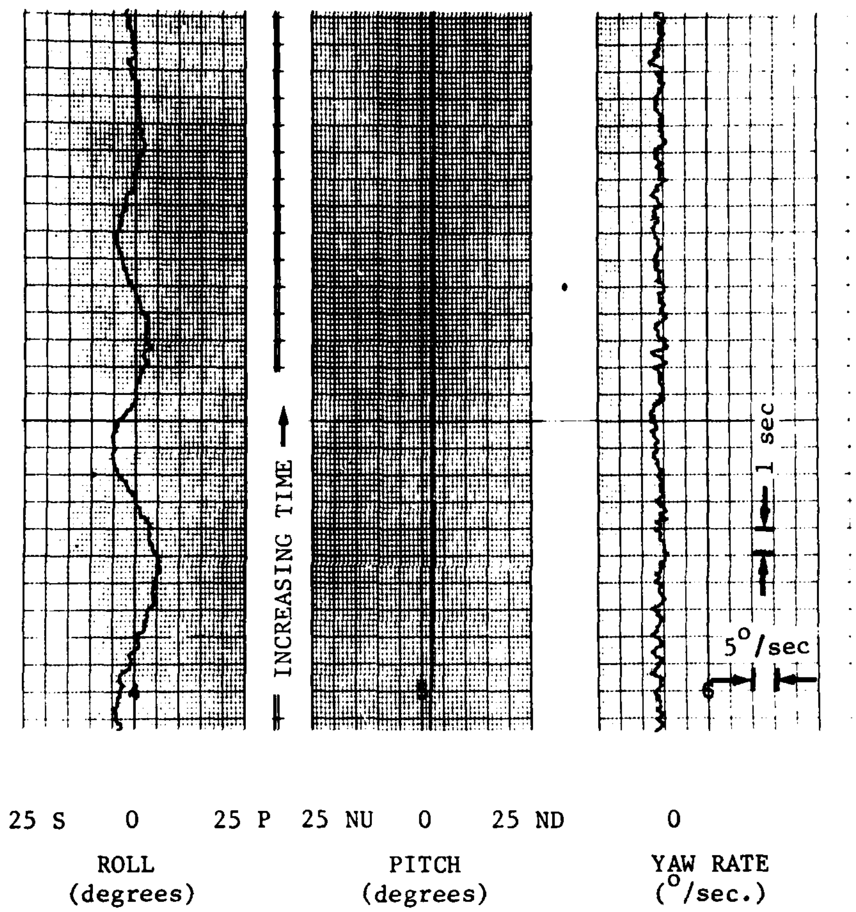
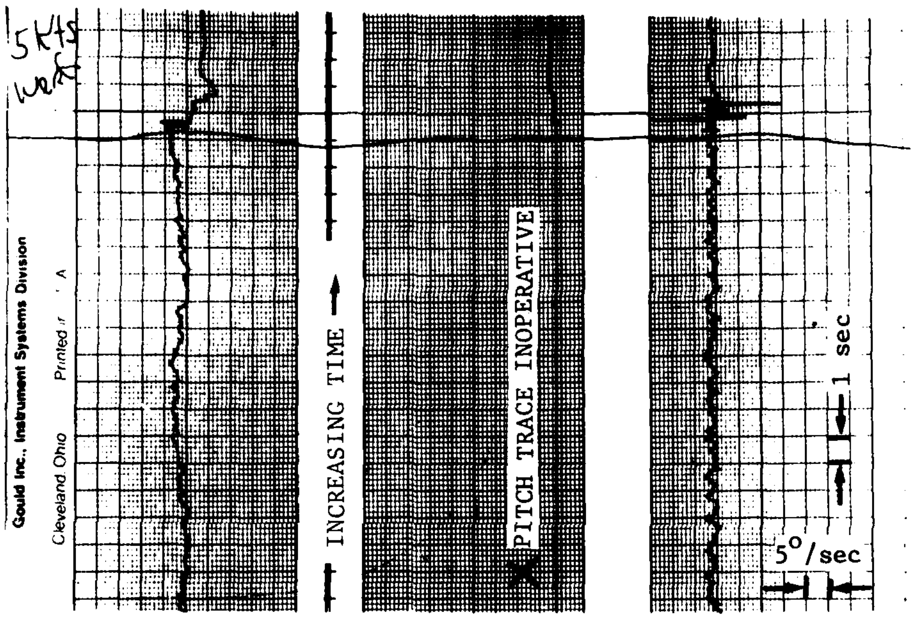


Figure 8b - At 15-Knot Towing Speed

Figure 8 (Continued)



25 S 0 25 P 25 NU 0 25 ND 0

ROLL (degrees) PITCH (degrees) YAW RATE (°/sec.)

Figure 8c - At 18-Knot Towing Speed

Figure 8 (Continued)

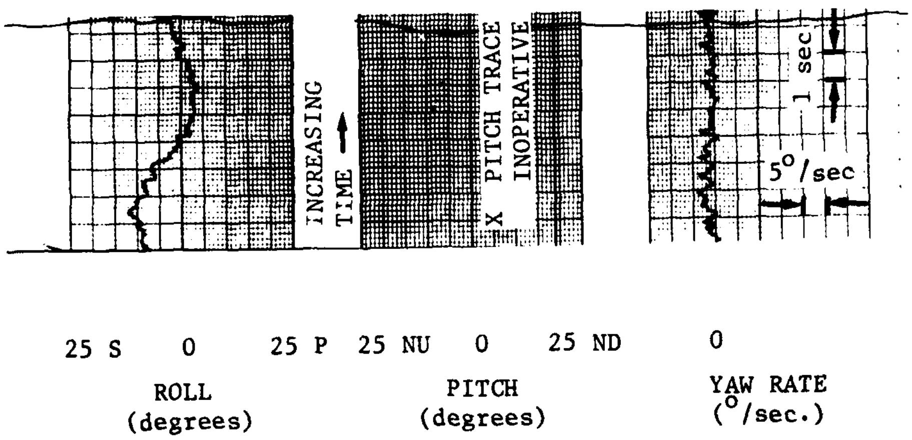


Figure 8d - At 20-Knot Towing Speed

Configuration 10, which consisted of one-half of the wooden sleeve attached to the underside of the model, the bow fin mounting bracket and the small vertical stabilizer extension produced an excellent, controllable tow but with a 14° nose-down pitch. Adding the upper sleeves produced no improvement. Full span horizontal trim tabs were attached to the upper horizontal fin (configuration 16). This produced excellent results. The pitch decreased to 8° nose-down, the towline tension decreased, and a very steady tow (the best to that time) was observed. The results are shown in Figure 9.

The reason for the superior performance of the wooden half-sleeve, as opposed to using both sleeves, is believed due to loss of effectiveness of the upper horizontal stabilizer as a result of the very large expansion of the lines of the after-body when the upper wooden half-sleeve was attached, as illustrated by Figure 3.

To reduce the suspected interference two sets of sleeves were fabricated from fiberglass. These are called "large tapered" and "small tapered"; the sizes are given in Figure 3. The small tapered sleeves produced excellent results. The best results were obtained with configuration 26 for which the V-fin was mounted on the underside of the body with the V inverted (Λ). The results are shown in Figure 10.

Configuration 27, which was the same as configuration 26 except that the V-fin was mounted on the upper side of the body produced results almost as good as those obtained with configuration 26. Configuration 26 was disturbed severely during the 18-knot run by yanking the towline (Figure 10c). The body in this configuration recovered immediately and gave all appearances of having high damping in all modes of the motion.

The body in configuration 26 towed with a 3° to 4° nose-down trim. The static (zero speed) trim was 7° to 8° nose-down, indicating that the center of weight in water was about 1/2-in. (1.3 cm) forward of the tow point, 2-1/2 in. (6.4 cm) full scale. With a 3° nose-down towing

Figure 9 - HYTOW Towing Characteristics with Wooden Half-Sleeves, Small Vertical Fin, Boundary Layer Trip and Horizontal Fin Trimming Flaps (Configuration 10)

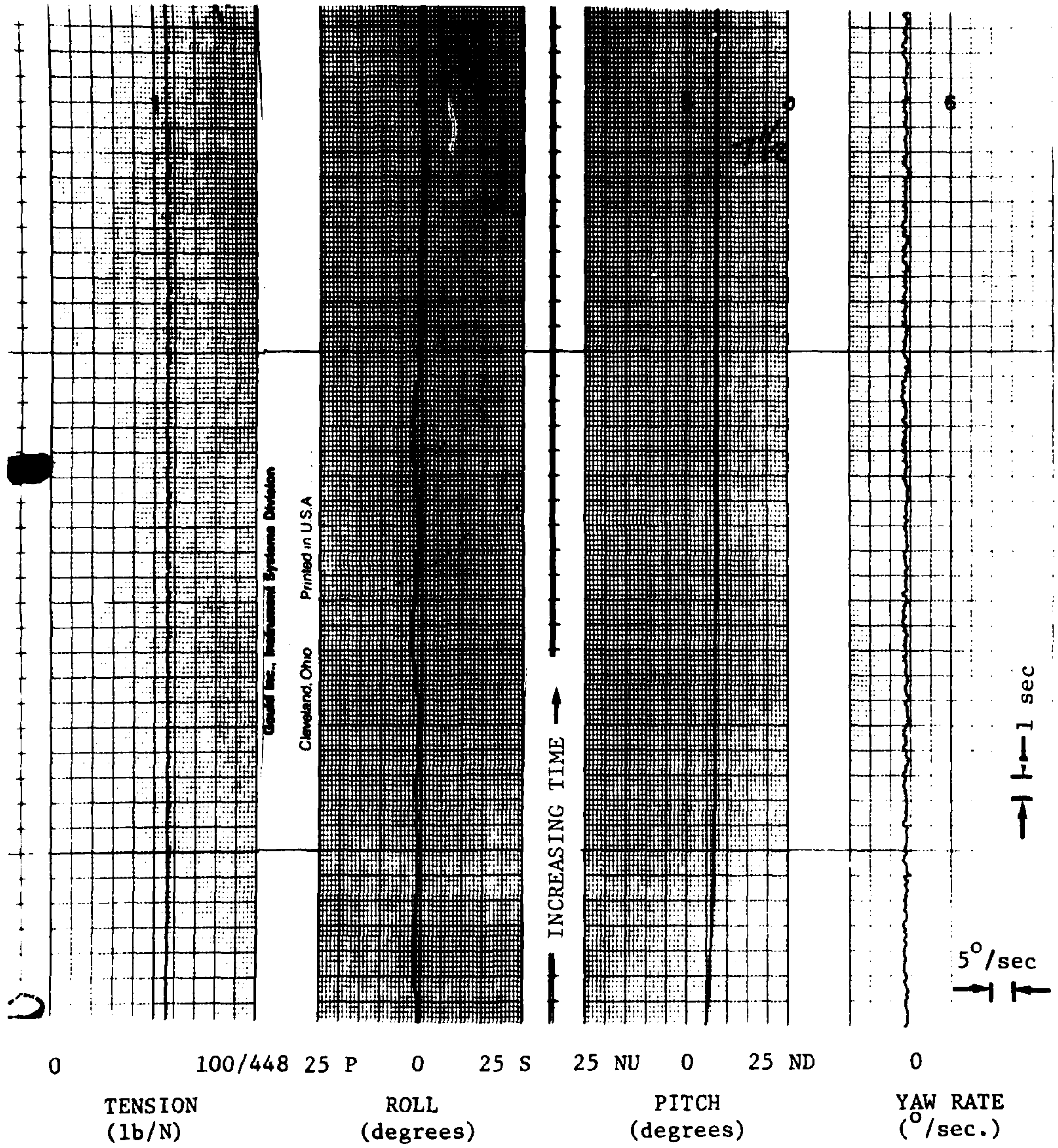


Figure 9a - At 11-Knot Towing Speed

Figure 9 (Continued)

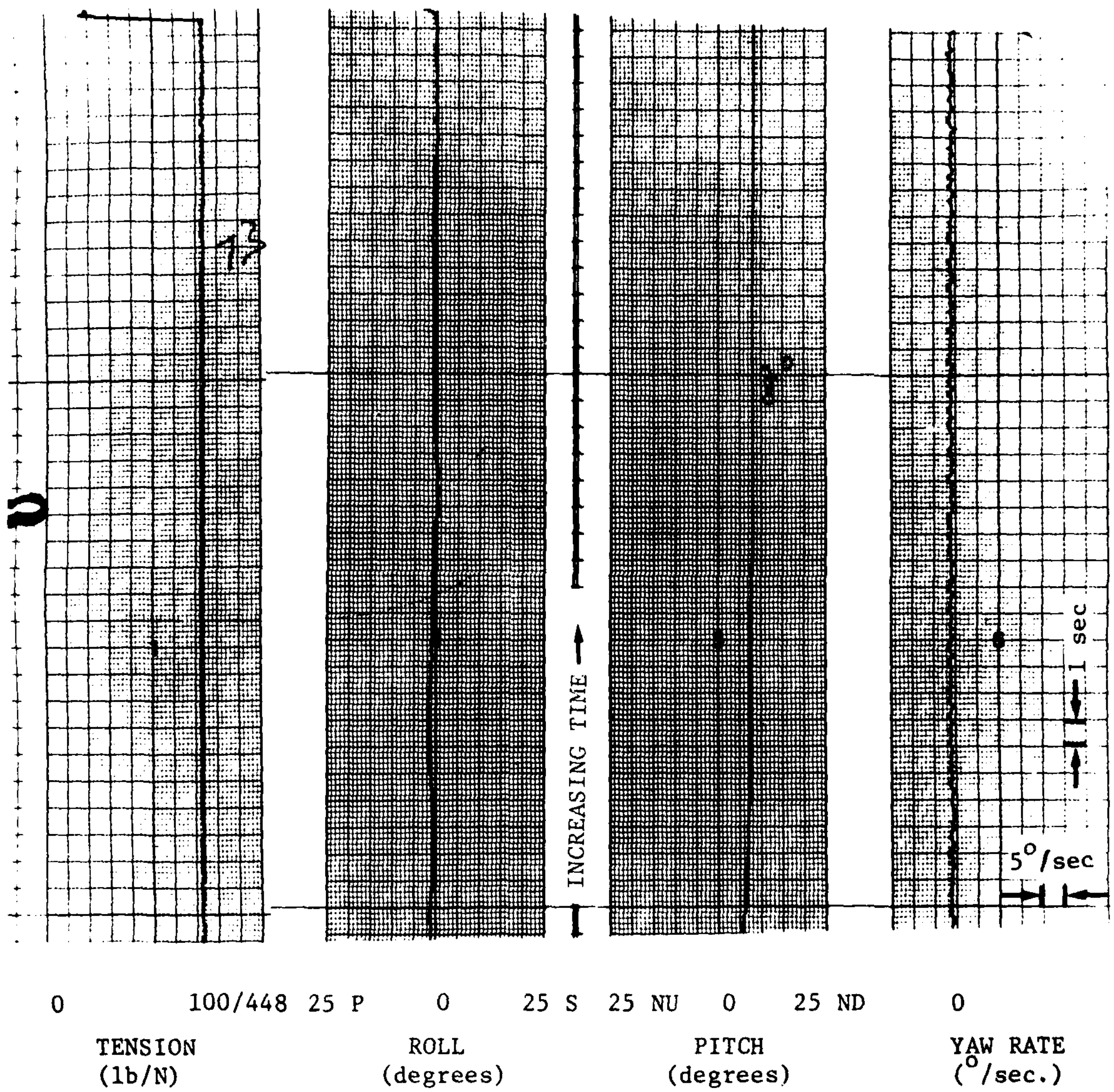


Figure 9b - At 13-Knot Towing Speed

Figure 9 (Continued)

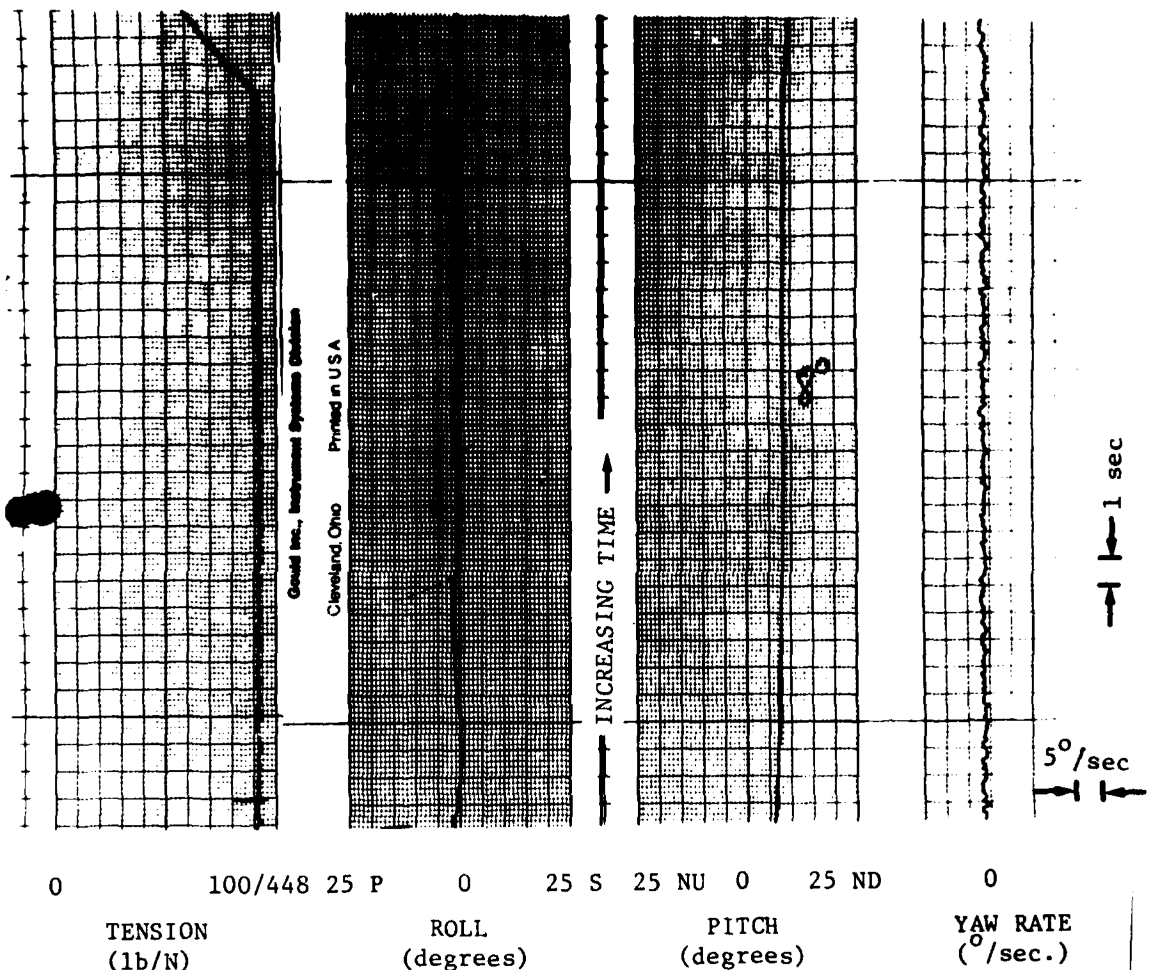


Figure 9c - At 15-Knot Towing Speed

Figure 9 (Continued)

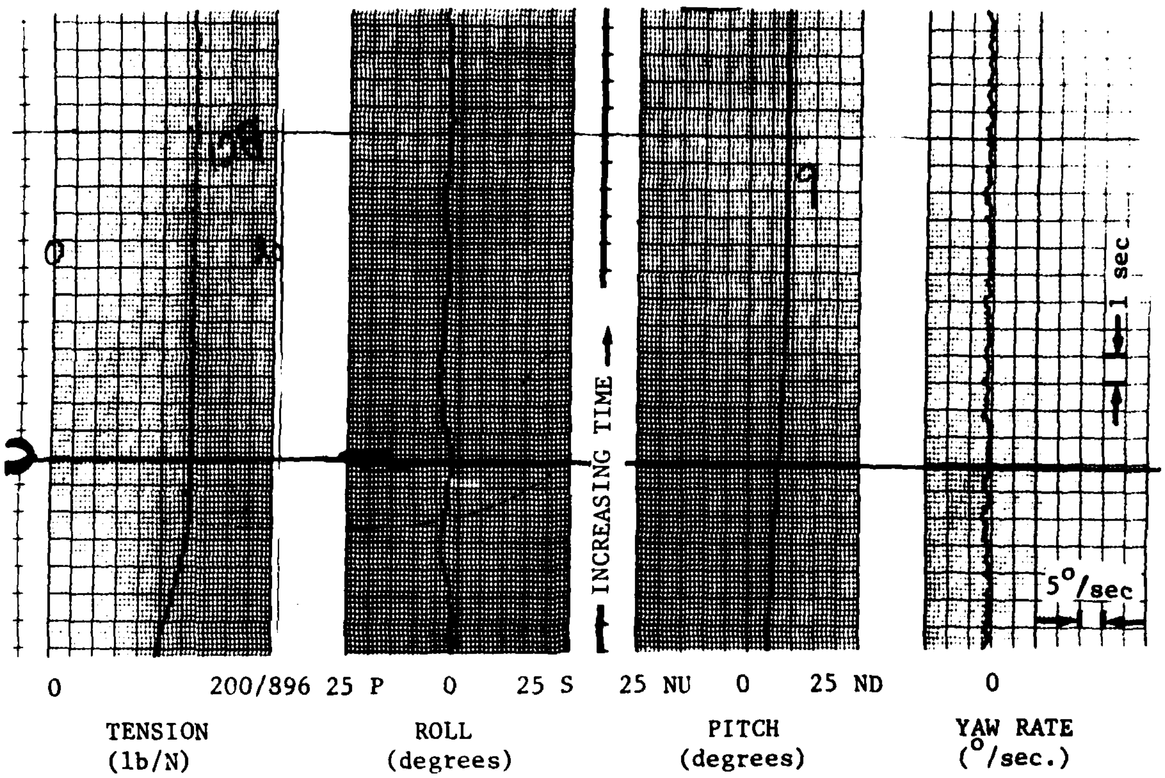


Figure 9d - At 18-Knot Towing Speed

Figure 9 (Continued)

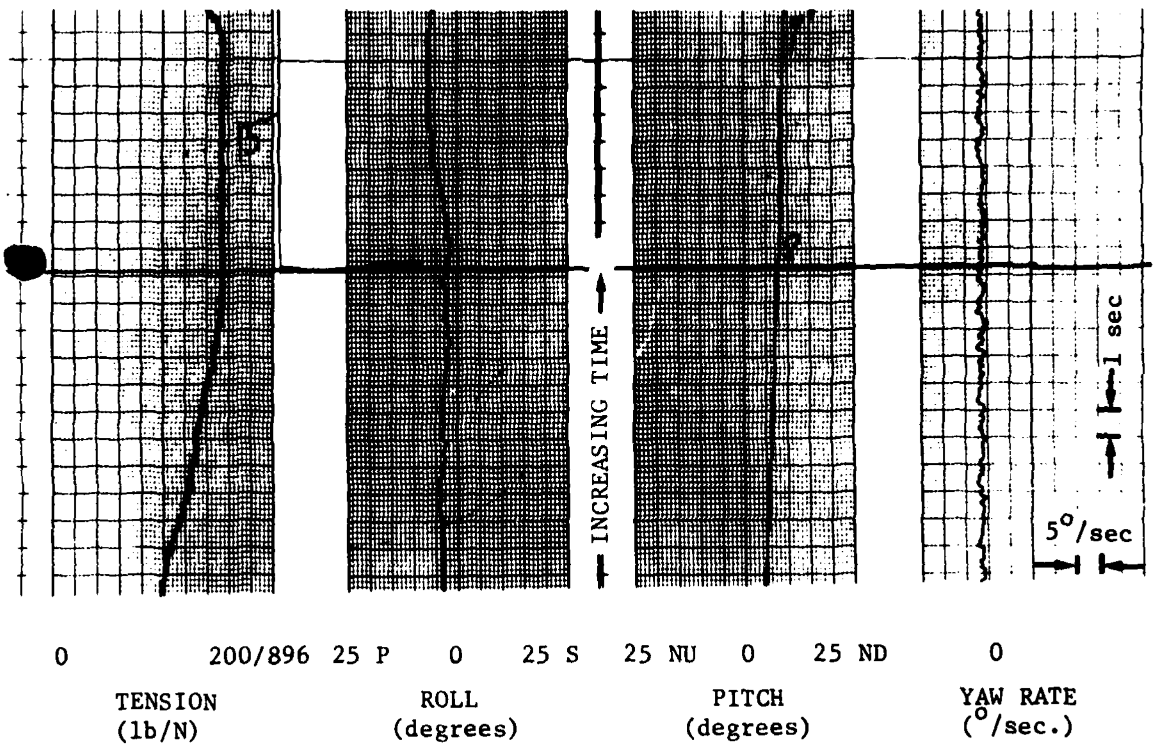


Figure 9e - At 20-Knot Towing Speed

Figure 10 - HYTOW Towing Characteristics with Small Tapered Sleeves, Inverted V-Tail and Boundary Layer Trip (Configuration 26)

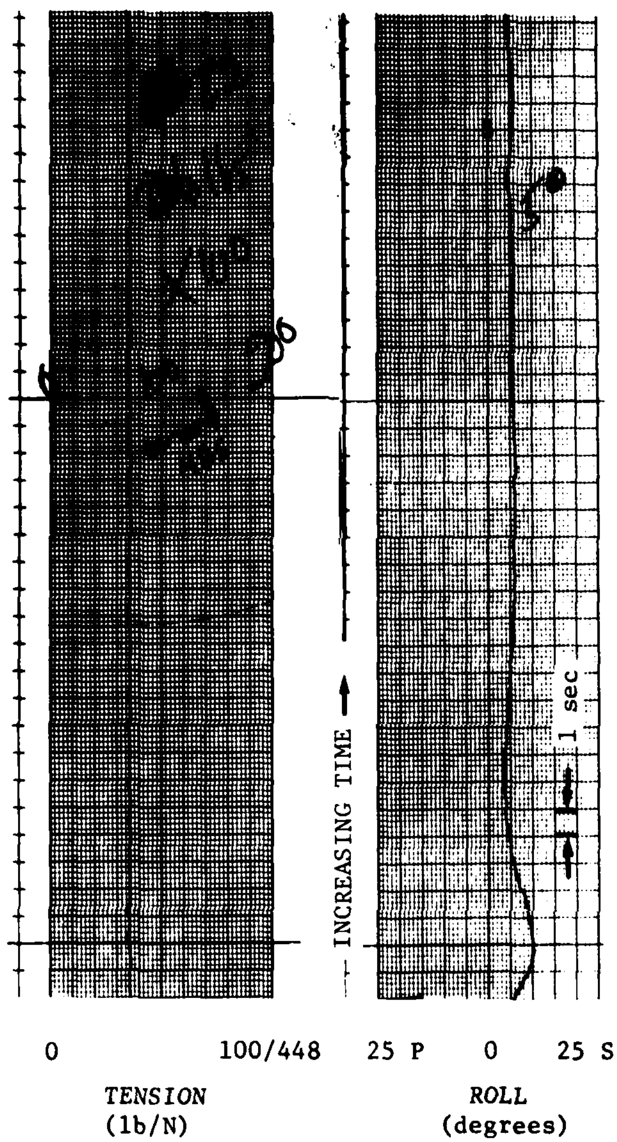


Figure 10a - At 13-Knot Towing Speed

Figure 10 (Continued)

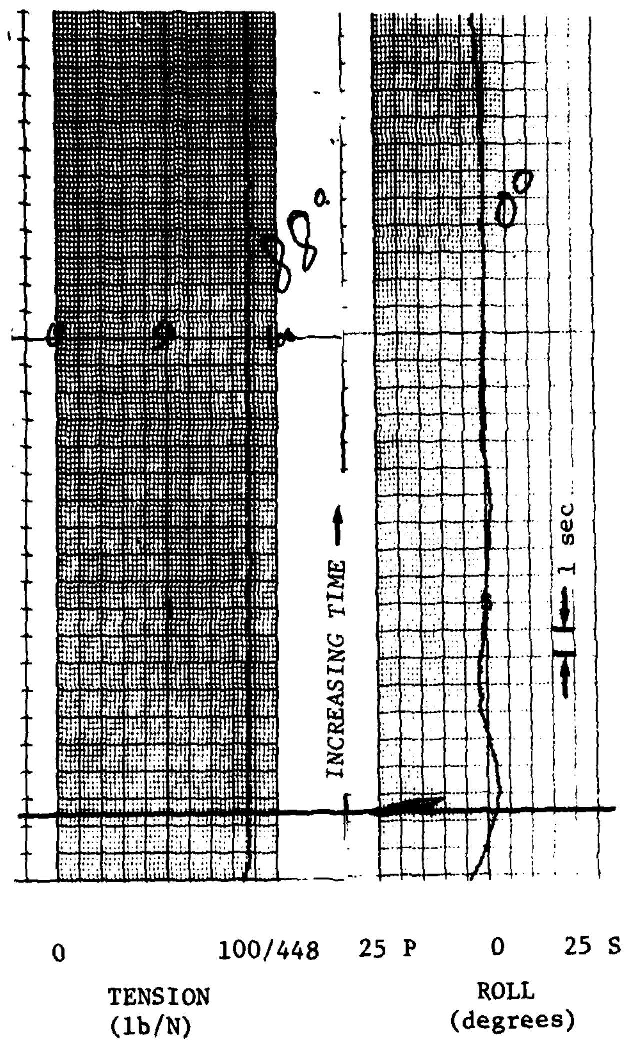


Figure 10b - At 15-Knot Towing Speed

Figure 10 (Continued)

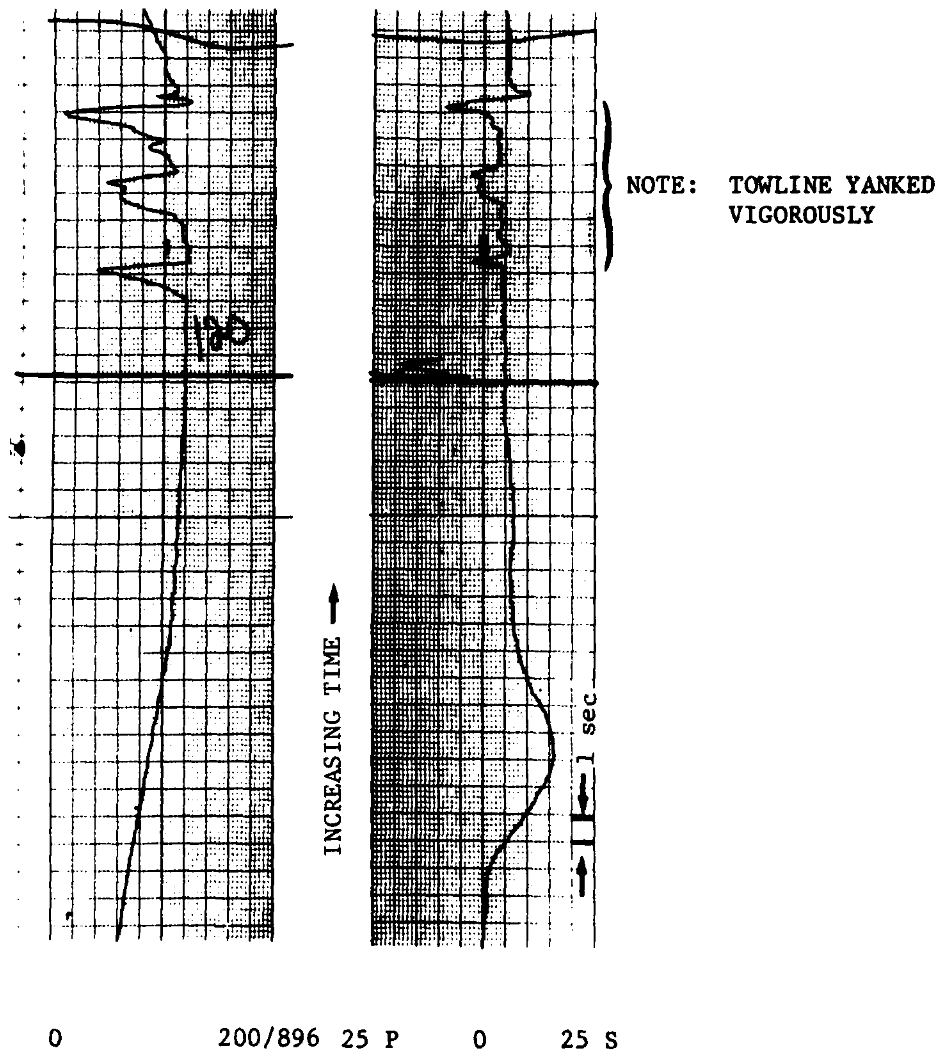


Figure 10c - At 18-Knot Towing Speed
(Body Disturbed by Pulling and Releasing the Towline)

Figure 10 (Continued)

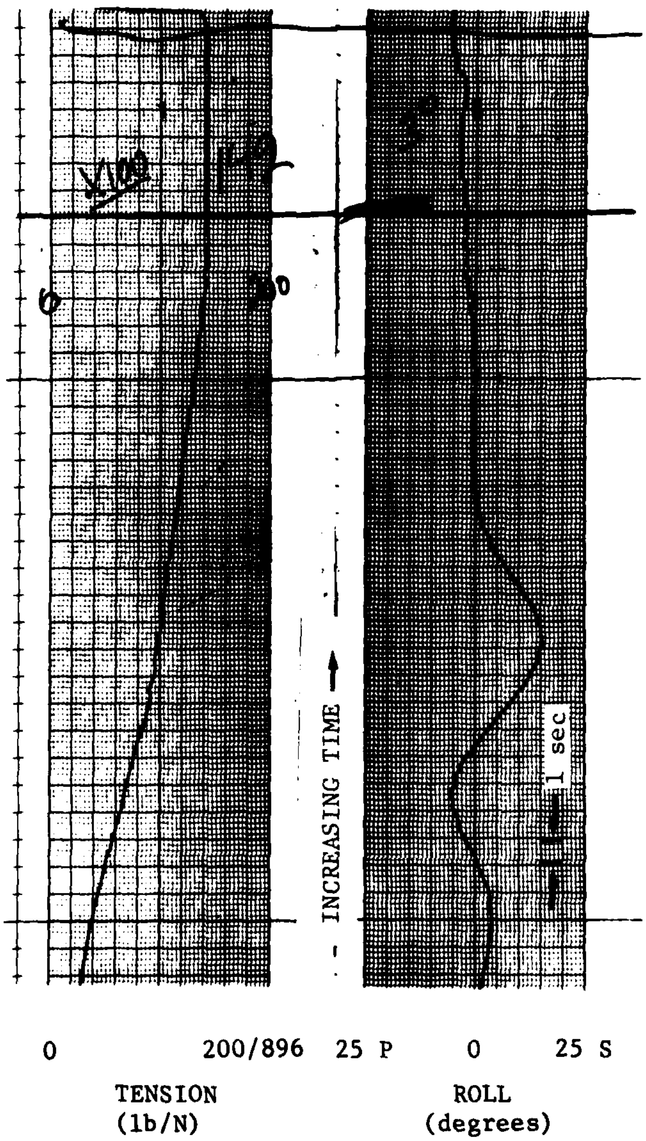


Figure 10d - At 20-Knot Towing Speed

attitude, this placed the center of weight in water about 1/3-in. (0.9 cm) ahead of the tow point when under tow.

CONCLUSIONS

The initial exploratory tows strongly inferred that the body suffered a turbulent flow separation. The random nature of the body's roll and sway motions indicated that the boundary of the area of separation changed in a random manner, giving rise to changes in flow circulation and hence to randomly varying forces and moments. However, based on the evidence available from the full-scale trials and the towed model results, determination that the full-scale failures resulted from a lateral instability or severe unsteadiness from turbulent flow separation is not possible. Modifications to the body to fix the boundary of the turbulent separation may have modified lateral stability characteristics as well. The model configured to match the parameters of the full-scale housing exhibited motions quite similar to those observed full-scale, but it did not broach even at the full-scale equivalent speed of 45 knots. This may be due to the fact that the towline used to tow the model was not precisely scaled, and in fact had a coefficient of drag about 2-1/2 times that of the prototype towline. This affects the lateral stability faborably; thus it can be concluded that from the standpoint of lateral stability, the model results are at least mildly nonconservative.

The most effective mechanism for reducing the low frequency unsteadiness consisted of enlarging the model diameter aft of the 70% L station with termination in a bluff circular base. Placing the V-fin on the bottom (inverted) gave further modest but noticeable benefits. Adding vertical fin area on the underside of the body appeared to produce additional benefits by increasing directional stability and yaw damping. It may be inferred from the pitch data that no significant change in drag results from the addition of the small sleeves. The model in prototype configuration (configuration 20) towed with a nose-down pitch attitude of about 5 degrees. This is in agreement with data

from the prototype. The modified body (configuration 26) towed with a 3 to 4 degree nose-down trim. It is speculated that the area of the flow separation on the unmodified body is about the same as the area of the base of the small sleeves used to fix the flow separation. Hence, no significant change in drag is to be expected.

RECOMMENDATIONS

It is recommended that the full-scale body be evaluated with the following modifications:

1. Necessary (Minimum) Modifications

   a. Expand the body coordinates aft of about the 70% L station to match the configuration identified as "small tapered sleeves" in Figure 3.

   b. Add a lateral (yaw) trim tab having minimum dimensions scaled-up from those shown in Figure 5 (5 in. (12.7 cm) x 7.5 in. (19 cm)).

   c. Within the limits imposed by existing handling/winching/launching/retrieving systems, make the body as heavy in water as possible.

   d. With due consideration to other constraints, ballast the body to place the center of weight in water as far below and forward of the tow point as feasible. One constraint to consider is the speed at which the body will assume an acceptable nose-down attitude when under tow. For the forward ballast location to be effective, the pitch when under tow (at the higher speeds) must be less than the zero-speed pitch.

2. Highly Desirable Modifications

   a. Interchange the V-fin presently on the upper side of the prototype body with the flat horizontal fin on the bottom, with the "V" inverted (Λ).

b. Arrange for remote control of the yaw trim tab. This will require a readout of the roll attitude (necessary in any event). This feature will allow rapid trimming when towing speed is changed.

3. Desirable

a. Add a vertical fin extension to the lower vertical skeg and move the horizontal stabilizer to the bottom of this fin. A scaled-up version of the fin identified as the "short fin" in Figure 4 is recommended. The horizontal tail (preferably the "Λ"-fin) should be set at a 2° angle of incidence, nose-down, the same as exists on the prototype as presently configured.

b. Make the yaw trimming device active, but with a very long time constant so that it only affects the equilibrium roll attitude of the body and does not enter into the dynamic response of the housing.

It is recommended further that the initial tows of the modified body be conducted on maximum cable scope and that tests be conducted up to the maximum towing speed while on straight course and at steady speed with minimum accleration of the hydrofoil to achieve speed changes.

Consideration should be given to performing the evaluations by towing off a weak link <u>at the ship</u> by means of a cable clamp, with an attached buoy to suspend and mark the body in case of loss. This will require extra preparation and stripping the bitter end of the tow cable from the winch. A separate electrical stub, with break away provisions, would be needed to permit towing performance to be monitored.

Space and weight available for equipment on warships are typically severely limited. The developer of towed sonar systems is thus under strong pressure to use towed bodies of minimum fineness ratio. But little literature exists on the characteristics of bodies in the range of fineness ratios greater than unity and less than about five. Further, no systematic investigation has been undertaken of methods for reducing

the spectral energy of the hydrodynamic forces and moments acting on bluff bodies. The results of these tests indicate that most of the conventional techniques may not work in the face of the extremely adverse pressure gradients typical of very bluff configurations. Moreover, the method recommended for stabilizing the HYTOW-type body results from an investigation having a very limited scope, and may be far from optimum, with respect to drag in particular.

It is therefore suggested, before extending these results to other than the housings to be used for the HYTOW hydrofoil evaluations, that tests be conducted in a large, low-turbulence facility equipped to provide for both force measurements and flow visualization to evaluate the efficacy of the recommended modifications and/or to improve thereon. It is believed that a program for general investigation of bluff-body flow would yield substantial benefits to future developments of towed systems.

ACKNOWLEDGEMENTS

The assistance of the Canadian Defense Research Establishment in providing the model of the HYTOW and other important information and data is gratefully recognized. The author also wishes to express his appreciation for the many technical suggestions advanced by Mr. William O'Neill.

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APPENDIX A
DYNAMIC SIMILARITY OF TOWED BODIES

DYNAMICALLY (FROUDE) SCALED BODIES

Geometrically similar bodies are said to be dynamically similar (in the sense that they undergo similar motions in similar times[4]) if the ratios of force-to-mass and moment (torque)-to-inertia are the same for both bodies. According to Duncan[4], for bodies whose linear dimensions are in the ratio λ, the condition is satisfied if the mass densities of the bodies are the same and the mass is distributed in the same geometric relation. The mass distributions need not be identical provided that the total mass, the centroids of mass and radii of gyration are in the proper ratios.

These ratios for immersed bodies one and two are given by Duncan[4] as:

Space scale*	$\beta = X_2/X_1 = \lambda$	
Time scale	$t = T_2/T_1 = \sqrt{\lambda}$	
Velocity scale	$u = U_2/U_1 = \sqrt{\lambda}$	
Linear acceleration scale	$\ddot{x} = \ddot{X}_2/\ddot{X}_1 = 1$	
Angular acceleration scale	$\ddot{\theta} = \ddot{\Theta}_2/\ddot{\Theta}_1 = t^{-2} = \lambda^{-1}$	
Body linear scale	$\ell = L_2/L_1 = \lambda$	
Body area scale	$s = S_2/S_1 = \lambda^2$	(A.1)
Body volume scale	$\forall = \forall_2/\forall_1 = \lambda^3$	
Body orientation scale	$\theta = \Theta_2/\Theta_1 = 1$	
Body moment of inertia scale	$n = N_2/N_1 = \lambda^5$	
Force scale	$f = F_2/F_1 = \lambda^3$	
Weight scale	$w = W_2/W_1 = \lambda^3$	

*The ratio of the locations of identical reference points in the bodies in the motion reference frame as distinguished from the ratio ℓ measured in a body reference frame.

For the above results to hold for immersed bodies, the hydrodynamic coefficients of the two bodies must be identical; i.e., Reynolds effects must be small. Bodies scaled in accordance with the above are said to accord with Froude scaling.

WEIGHT AND MASS

A body immersed in a fluid of density ρ is subject to a buoyancy force B given by

$$B = W_v - W$$

where W is the weight in water and W_v is the weight in a vacuum. We define the submerged specific gravity as Δ, where

$$\Delta = W/ \mathcal{V}g$$

g being the acceleration of gravity and $\mathcal{V}$ the volume enclosed by the outer surface of the body.

The total mass of the body M is given by

$$\underline{M} = M_v + M_w + M_A$$

where M_v stands for the mass of the structure, M_w the mass of the water (or fluid) contained inside the body, and M_A the added hydrodynamic mass. The mass of the contained fluid is given by

$$M_w = \rho(\mathcal{V} - \mathcal{V}_s)$$

where $\mathcal{V}$ is the envelope volume and $\mathcal{V}_s$ is the volume of the structure. But

$$\mathcal{V}_s = B/\rho g$$

therefore

$$\underline{M} = \frac{W_v}{g} + \rho \left[\mathcal{V} - \frac{W_v - W}{\rho g} \right] + M_A$$

$$\underline{M} = \rho\forall + \frac{W}{g} + M_A$$

We may write, as is the usual practice,

$$M_A = k\rho\forall$$

where k is the added mass factor[5]. Then $\underline{M}$ may be put in the form

$$\underline{M} = (1 + \Delta + k)\ \rho\forall$$

or $$\underline{M} = M\rho\forall$$

by defining $$M = 1 + \Delta + k$$

The ratio of body force to mass is thus

$$f_{m_b} = g\Delta/(1 + k + \Delta) = g\Delta/M.$$

Let $\Delta_2/\Delta_1 = \delta$ and $M_2/M_1 = m$. The ratio $(f_{m_b})_2/(f_{m_b})_1 = f_b$ is then,

$$f_b = \delta/m. \tag{A.2}$$

Clearly $M_2 = \lambda^3 M_1$ and $w = \lambda^3$ if $\delta = m = 1$.

In the case $\delta \neq 1$; $m \neq 1$, $w \neq \lambda^3$, and $f \neq \lambda^3$, the scaling ratios presented earlier do not hold. It is therefore doubtful if the dynamic motions of two bodies may be scaled under these conditions.

NON-FROUDE SCALED BODIES

If $\overline{X}$ is the vector locating a reference point in a towed body (assumed here to be the towpoint) relative to a coordinate frame fixed

in space, T is the time and $\overline{\Theta}$ the rotation of the body relative to the coordinate frame, we may write

$$d^2\overline{X}/dT^2 = \ddot{\overline{X}} = \overline{F}_b/M + \overline{F}_h/M + \overline{F}_e/M \tag{A.3}$$

$$d^2\overline{\Theta}/dT^2 = \ddot{\overline{\Theta}} = \overline{H}_b/I + \overline{H}_h/I \tag{A.4}$$

where respectively, $\overline{F}_b$ and $\overline{H}_b$ are the body force and moment, $\overline{F}_h$ and $\overline{H}_h$ are the hydrodynamic force and moment acting on the body, $\overline{F}_e$ is an external force due to the towline, and I is the moment of inertia.

If for two bodies having the geometric scale λ,

$$\ddot{\overline{X}}_2 = p\ddot{\overline{X}}_1$$

and

$$\ddot{\overline{\Theta}}_2 = q\ddot{\overline{\Theta}}_1$$

we require

$$(\overline{F}_b/\underline{M})_2 = p(\overline{F}_b/\underline{M})_1$$

$$(F_h/\underline{M})_2 = p(\overline{F}_h/\underline{M})_1$$

etc.

and

$$(\overline{H}_b/I)_2 = q(\overline{H}_b/I)_1$$

$$(\overline{H}_h/I)_2 = q(\overline{H}_h/I)_1$$

etc.

as then

$$\frac{\ddot{\overline{X}}_2}{\ddot{\overline{X}}_1} = \frac{p(\overline{F}_b/\underline{M})_1 + p(\overline{F}_h/\underline{M})_1 + p(\overline{F}_e/\underline{M})_1}{(\overline{F}_b/\underline{M})_1 + (\overline{F}_h/\underline{M})_1 + (\overline{F}_e/\underline{M})_1} = p$$

and similarly, $\ddot{\overline{\Theta}}_2/\ddot{\overline{\Theta}}_1 = q$.

Space and Time Scales

As defined earlier, let $\beta = \overline{X}_2/\overline{X}_1$ and $t = T_2/T_1$. It follows from the definition of the derivative that

$$\ddot{\overline{X}}_2/\ddot{\overline{X}}_1 = \beta/t^2 = p$$

But $\beta/t^2 = (\overline{F}_b/\underline{M})_2/(\overline{F}_b/\underline{M})_1 = f_b$. From equation A.2

$$\beta/t^2 = \delta/m = p.$$

The inertia tensor can be thought of as forming a surface about the origin of body coordinates whose normalized radial distance is the sum of $1 + \Delta$ (thus a sphere) and the radial distance to the surface of the ellipsoid whose semiaxes in body coordinates are k_a, k_b and k_c. The latter can be put in the form

$$r = k_a k_b k_c F(\Omega)$$

where Ω is the compound angle between a principal axis and the direction of $\overline{X}$. For a two dimensional body, e.g., $r = k_a k_b/(k_a{}^2\sin^2\Omega - k_b{}^2\cos^2\Omega)$. If $\Omega(T_2) = \Omega(T_1)$ the ratio m will be constant although both M_2 and M_1 are functions of $\Omega(t)$. Thus m can be put in the form

$$m = \frac{1 + \delta\Delta_1 + r(\Omega)}{1 + \Delta_1 + r(\Omega)}$$

The hydrodynamic terms consist of force and damping terms. The force term (now neglecting vector notation) can be put in the form

$$F_h = c_\alpha \alpha \rho S U^2/2$$

where c_α is the force coefficient and α the angle of attack. Assuming $\alpha(T_2) = \alpha(T_1)$

$$f_h = (F_h/\underline{M})_2/(F_h/\underline{M})_1 = u^2/\lambda m = p$$

(where λ is, of course, the body scale ratio). Thus, the ratio of velocities must be

$$u = \sqrt{p\lambda m} = \sqrt{\delta\lambda}$$

to satisfy the requirement $\ddot{x} = p$. But

$$u = \beta/t,$$

which with the earlier result, $\beta/t^2 = \delta/m$, gives

$$\beta = m\lambda$$

and

$$t = m\sqrt{\lambda/\delta}$$

Recall that F_e is an externally applied force. Let $f = F_{e_2}/F_{e_1}$, then

$$f_e = f\underline{M}_1/\underline{M}_2 = f/m\lambda^3 = \delta/m$$

Therefore

$$f = \delta\lambda^3$$

which holds for all force ratios.

The damping term can be written in the form

$$F_h = R\ dU/U$$

where R is a resistance (i.e. a force). Since dU/U is dimensionless,

$$(R/\underline{M})_2/(R/\underline{M})_1 = \delta\lambda^3/m\lambda^3 = \delta/m = p.$$

Therefore dynamic similarity exists for rectilinear motions of bodies having apparent specific gravity ratios δ and mass ratios m at the speed ratio $u = \sqrt{\delta\lambda}$, for which the space scale $\beta = m\lambda$ at times scaled according to $t = m\sqrt{\lambda/\delta} = \sqrt{m\beta/\delta}$.

Rotational Motion

The force coefficients are a function of body attitude relative to the flow, thus for the ratios f to hold, it is necessary that

$$\Theta_2(T_2) = \Theta_1(T_1).$$

The angle of attack is given by

$$\alpha = \Theta - Z,$$

where Z is the angle of the tangent to the trajectory relative to the fixed reference frame. Let x and y be coordinates of the fixed reference frame, then

$$Z = \tan^{-1} y/x$$

But $\qquad y_2 = \beta y_1$ and $x_2 = \beta x_1$,

therefore

$$Z_2 = \tan^{-1} y_2/x_2 = \tan y_1/x_1 = Z_1$$

However, for $\Theta_2(T_2) = \Theta_1(T_1)$, it is necessary that

$$\ddot{\Theta}_2/\ddot{\Theta}_1 = 1/t^2 = \delta/(m^2\lambda) = \delta/(m\beta) \qquad \text{(A.5)}$$

Thus, in equation A.4, it is necessary that a parameter q (applied in the same manner as p) be equal to $\delta/(m\beta)$.

The body moment due to W can be written in the form

$$H_b = AW \sin(\sigma + \Theta)$$

where σ is an angle relative to a body reference line and the directrix to A, where A is the distance from the towpoint to W. At zero speed, $H_b = 0$. Therefore $\Theta = \Theta_0 = -\sigma$. Hence, the direction line to W must form the same angle relative to a body reference in both model and prototype. We now write

$$h_b = (H_b/I)_2/(H_b/I)_1$$

$$= \frac{A_2W_2}{A_1W_1}\,\frac{M_1N_1^2\rho\forall_1}{M_2N_2^2\rho\forall_2} = \frac{a\delta}{mn^2}$$

where $I = MN^2\rho\forall$, N being the radius of gyration and $A_2/A_1 = a$.

From equation A.4,

$$h_b = \frac{a\delta}{mn^2} = \frac{\delta}{m\beta}$$

or,

$$a/n^2 = \beta^{-1} \tag{A.6}$$

A second component of body moment is given by

$$H_b = M\rho\forall A\ddot{X}$$

Applying the now familiar technique of taking ratios;

$$h_b = \frac{a\beta}{n^2t^2} = \frac{1}{t^2}$$

Thus,

$$a/n^2 = \beta^{-1}$$

in agreement with equation (A.6).

The moment (or torque) due to orientation to the flow (angle of attack) may be put in the form of the product of a force and a lever arm, i.e.,

$$H(\alpha) = F(\alpha)\alpha L$$

$$\therefore \quad h_h(\alpha) = f_h(\alpha)\frac{\lambda}{n^2}$$

$$= \frac{u^2}{\lambda}\frac{\lambda}{mn^2} = \frac{\delta}{m\beta}$$

and

$$n^2 = \beta\lambda = m\lambda^2 \qquad (A.7)$$

Thus, using equation (A.6)

$$a = \lambda \qquad (A.8)$$

That is, W must have the same relative location in model and prototype.

The rotary damping term in H_h is of the form

$$F(\alpha)L^2(\dot{\theta} - \dot{Z})/U$$

Again taking ratios;

$$h_h(\alpha) = \frac{p\lambda^2}{\beta mn^2} = \frac{\delta}{m\beta}$$

This ratio is satisfied only if

$$n^2 = \lambda^2/m \qquad (A.9)$$

Thus, recalling equation (A.7), it is not possible to properly scale the radius of gyration to simultaneously satisfy the rotary damping and moment due to orientation relative to the fluid.

DISCUSSION

The results developed indicate that an immersed body can be properly dynamically scaled only under the condition that the scale of motion in space and the body length scale are identical, i.e., $\beta = \lambda$, which requires $\delta = m = 1$.

It is possible, for the case in which the ratio of the specific gravity, δ, of two bodies is not unity to satisfy all scaling criteria excepting the rotary-damping to mass ratio. If $\Delta_2/\Delta_1 = \delta$ and $M_2/M_1 = m$, we find

$$\begin{aligned}
\beta &= m\lambda \\
t &= m\sqrt{\lambda/\delta} \\
u &= \sqrt{\delta\lambda} \\
f &= \delta\lambda^3 \\
a &= \lambda \\
n &= \sqrt{m}\cdot\lambda \\
\theta &= 1 \\
\sigma &= 1 \\
&\text{etc.}
\end{aligned} \tag{A.10}$$

However, in those cases where the rotary damping is small, equations A.10 may be applied with due caution.

The rotary damping depends on the term $L(\dot{\Theta} - \dot{Z})/U$. The ratio of the damping terms is $\lambda/\beta = m^{-1}$. Thus, if $\delta > 1$, i.e. body 2 is more dense than body 1, the rotary (pitch, yaw) damping of prototype motions will be less than that for model motions. Hence, it is conjectured that the limiting speed for stability of the prototype will be overstated in applying equation A.11 to the speed measured with the model. The reverse, of course, holds for $\delta < 1$.

Applying the definition of δ to the expression for the velocity ratio in equation A.9, we find

$$u = \sqrt{\frac{W_2 W_1}{\lambda^2}} \tag{A.11}$$

a form convenient for application. Computation of δ is also facilitated by noting

$$\delta = w/\lambda^3 \tag{A.12}$$

APPENDIX B

STEADY-STATE KITING OF TOWED BODIES

Asymmetry of a towed housing results in a side force (Y_0) and moment (N_0) when the angle of yaw (ψ) is zero. The resultant side force is the sum of Y_0 and the side force Y developed by the body at the yaw angle ψ corresponding to that for which the sum of the yawing moments about the towpoint vanishes. This produces a sway and by virtue of the towline constraint, a rolled attitude of the body.

Referring to Figure 11, for equilibrium, the steady-state value of the rolling moment about the towpoint must vanish, i.e.,

$$zY_0 + RY_\psi \psi - Wh\phi + L_\psi \psi = 0 \qquad \text{(B.1)}$$

and the yaw moment taken about a vertical line through the towpoint also must vanish:

$$N_0 + N_\psi \psi - W\phi a f(\alpha) = 0. \qquad \text{(B.2)}$$

In the equations above:

$f(\alpha)$	= $\cos\alpha - (h/a)\sin\alpha$
L	= the rolling moment introduced by a lifting surface, e.g., a wing
L_ψ	= $\partial L/\partial\psi$
N	= the hydrodynamic moment
N_ψ	= $\partial N/\partial\psi$
R	= the body radius in the lateral plane containing the towpoint
Y_ψ	= $\partial Y/\partial\psi$
z, h, a	= as defined in Figure 11
α	= the angle of attack
ϕ	= the roll angle

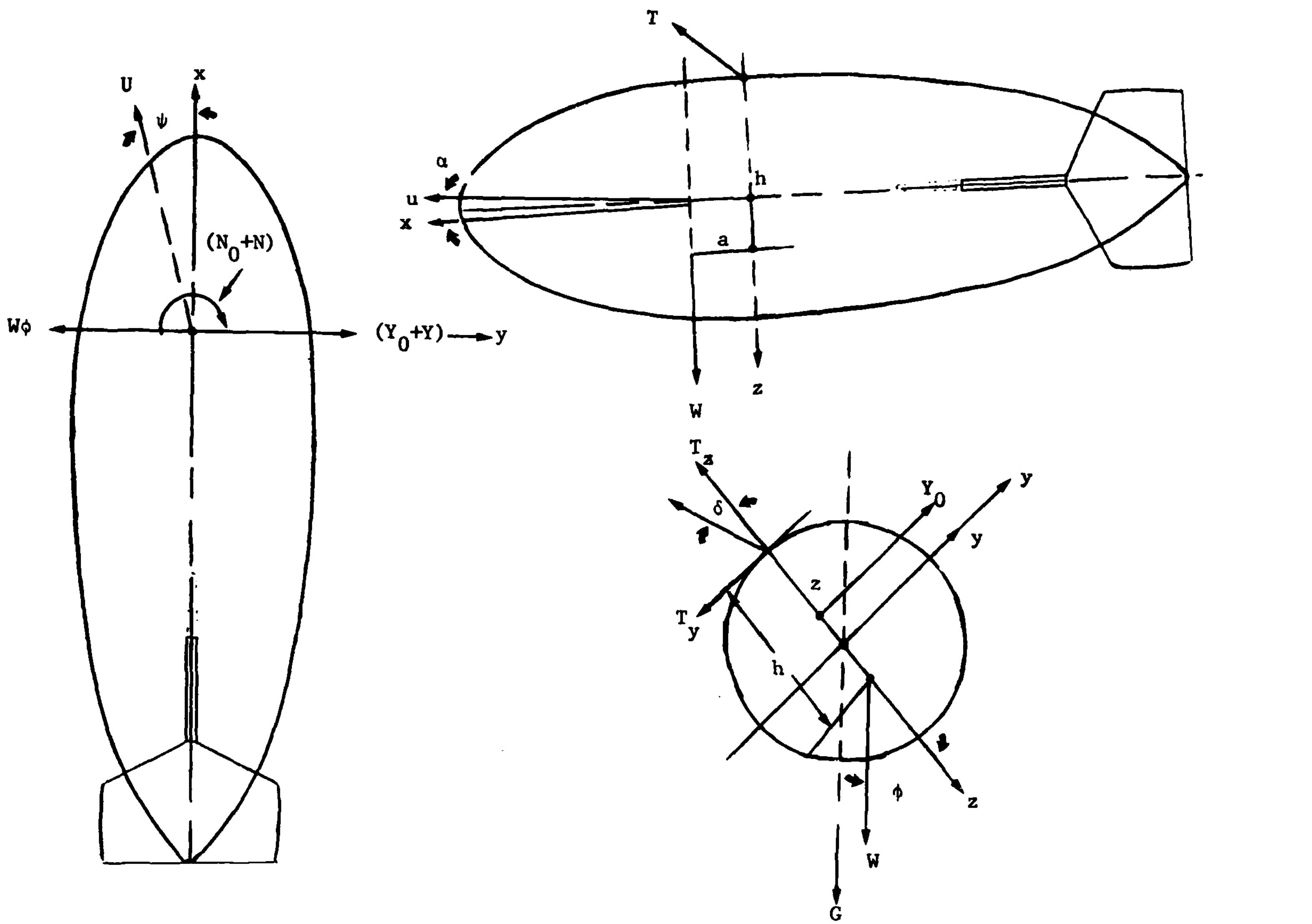

Figure 11 – Definition Sketch for Towed Body Kiting

Eliminating ψ between equations B.1 and B.2,

$$\phi = \frac{zY_0}{hW}\left[\frac{\dfrac{1}{RY_\psi + L_\psi} - \dfrac{N_0}{zY_0N_\psi}}{\dfrac{1}{RY_\psi + L_\psi} - \dfrac{\tan\alpha_0 f(\alpha)}{N_\psi}}\right] \tag{B.3}$$

where $\tan\alpha_0 = a/h$.

Setting

$$Y_\psi = C_{Y_\psi} qs$$

$$N_\psi = C_{N_\psi} qs\ell$$

$$Y_0 = C_{Y_0} qs$$

$$N_0 = C_{N_0} qs\ell$$

in equation B.3; it may be put in the form

$$\frac{\phi W}{C_{Y_0} qs}\,\frac{h}{z} = \left[\frac{1 - (\ell/z)(C_{N_0}/C_{Y_0})\ \ \bar{C}_{Y_\psi}/C_{N_\psi}}{1 - f(\alpha)\tan\alpha_0\ \ \bar{C}_{Y_\psi}/C_{N_\psi}}\right] \tag{B.4}$$

where $\bar{C}_{Y_\psi} = C_{Y_\psi} R/\ell + C_{L_\psi}$.

In general C_{L_ψ} will be positive as dynamic stability generally requires a wing with positive dihedral (in the aerodynamic sense). Also, no loss in generality results from setting $h = z = R$, and assuming the body to tow at zero angle of attack, $f(\alpha) = 1$. Note that if $\alpha = \alpha_0$, $f(\alpha) = 0$. With these substitutions, we may write

$$\frac{\phi W}{qs} = C_{Y_0} \left[\frac{1 - 2F(C_{N_0}/C_{Y_0})\overline{C}_{Y_\psi}/C_{N_\psi}}{1 - \tan\alpha_0 \overline{C}_{Y_\psi}/C_{N_\psi}} \right] \tag{B.5}$$

where the fineness ratio $F = \ell/2R$.

The ratio of the asymmetric force to the asymmetric moment may be negative, zero, or positive. The first case would correspond to a misaligned tail plane, for example, the second to the case where the side force due to asymmetry acts through a vertical line that passes through the towpoint, and the last to the case where the line of action is forward of the towpoint.

For a properly designed body $C_{N_\psi} < 0$. Assume $C_{N_0}/C_{Y_0} = -1$, i.e., case 1. Equation B.5 becomes

$$F(\phi) = \frac{\phi W}{C_{Y_0} qs} = \frac{1 - 2F\overline{C}_{Y_\psi}/|C_{N_\psi}|}{1 + \tan\alpha_0 \overline{C}_{Y_\psi}/|C_{N_\psi}|} \tag{B.6}$$

$\overline{C}_{Y_\psi}/|C_{N_\psi}| = 0$ represents a body with infinite directional stiffness. For this case $F(\phi) = 1$. Also, $F(\phi) = 0$ if $2F\overline{C}_{Y_\psi}/|C_{N_\psi}| = 1$ and approaches the value $-2F\text{ctn}\alpha_0$ as $|C_{N_\psi}| \to 0$.

For case 2, $C_{N_0} = 0$, and thus

$$F(\phi) = 1/(1 + \tan\alpha_0 \overline{C}_{Y_\psi}/|C_{N_\psi}|)$$

For the body with high yaw stiffness, $F(\phi) \to 1$. For small yaw stiffness, $F(\phi) \to 2F\text{ctn}\alpha_0 |C_{N_\psi}|/C_{Y_\psi}$.

Looking at case 3, (assuming $C_{N_0}/C_{Y_0} = +1$),

$$F(\phi) = \frac{1 + 2F\bar{C}_{Y_\psi}/|C_{N_\psi}|}{1 + \tan\alpha_0\bar{C}_{Y_\psi}/|C_{N_\psi}|} \tag{B.7}$$

For this case, $F(\phi) \to 1$ if the yaw stiffness is large and increases monotonically as the yaw stiffness is reduced and for the limit,

$$|C_{N_\psi}| \to 0, \qquad F(\phi) \to +\ 2F\mathrm{ctn}\alpha.$$

The values of the zero yaw side force and moment coefficients are not known beforehand. There is thus little to guide the designer in the above results, except to note that $\tan\alpha_0$ should be as large as possible, since if $|C_{N_\psi}|$ is small, $F(\phi) \to \pm 2F\mathrm{ctn}\alpha_0 C_{N_0}/C_{Y_0}$ in all cases, except where $C_{N_0} = 0$. Also, for all cases, $F(\phi) \to 1$ as $|C_{N_\psi}| \to \infty$. Therefore W should be as large as other constraints permit.

We may write

$$\phi = \frac{qs}{W} f(\psi,\psi_0).$$

If we assume $f(\psi,\psi_0)$ to be the same for two bodies having the linear scale ratio λ,

$$\phi_2/\phi_1 = (W_1/W_2)\lambda^2 U_2^2/U_1^2$$

and the roll angles will be the same at the speed ratio

$$u = \sqrt{\frac{W_2/W_1}{\lambda^2}} \tag{B.8}$$

which is the same as the result obtained from dynamic scaling considerations (equation A.11).

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